THE MECHANICAL DESIGN OF AN ELECTRIC TOWN CAR

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by

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SUMMARY

This report examines the mechanical design of the University of Canterbury Mark II Electric Town Car. An interconnected suspension system is proposed and analysed in some detail. The selection of suspension geometry to incorporate anti-dive and to restrict body roll is discussed and the rear suspension trailing arms are analysed. The general layout features a backbone chassis with batteries between the front wheels and beneath the front passengers, with the electric motors and rear suspension beneath the rearward facing rear passengers. Two alternative methods of construction are proposed and discussed, including the adaption of the A40 Farina bodywork.

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CHAPTER 1

INTRODUCTION

The requirement for an Electric Town Car has arisen as a result of the development, by Mr. D. J. Byers of the Electrical Engineering Department, of an electronic control system, using static inverters, and powering AC induction motors (Byers and Harman, 1975 and Byers, 1977).

At present an interim vehicle exists, which has been used as an electric test-bed vehicle and has proven the electric drive system to be satisfactory (Harman and Byers, 1980).

A practical saleable electric town car, to replace the interim vehicle and to confirm the performance of the electric drive system was required. Such a vehicle must have a practical minimum weight, a body design of low aerodynamic resistance, a comfortable ride and an efficient drive system. The concept therefore is a full four seater saloon car, aimed at the second car market. It should be comfortable and have no distracting features that would detract from the electric vehicle capacity.

The original object was that the vehicle be built from scratch, rather than be based on an existing internal combustion engine car. This allows for the proper location of batteries for easy access and/or quick exchange and allows the body to be designed to minimise rolling and windage losses.

It was envisaged that this vehicle might be a production prototype, so that methods of construction and other design considerations should keep this in mind.

The requirements of low mechanical rolling resistance and low aerodynamic drag for any vehicle and particularly electric vehicles has been documented (Byers and Harman, 1975, and Harman, 1977).

The use of toothed belt or chain drive, direct from the motors to the rear wheels eliminates mechanical losses that would otherwise exist in the differential, drive-shaft bearings etc. The differential effect is achieved electrically.

Special locally made 145-SR-13 radial ply tyres are utilized, incorporating features that decrease the energy losses and hence the rolling resistance. The tyres have two circumferential steel belts, one radial ply and lighter than normal sidewall construction.

Wind tunnel tests were conducted (MacDonald, 1978) aimed at reducing the drag coefficient of the electric town car. This was done by modifying a one-fifth scale model of a proposed body shape designed by the Wellington Polytechnic in collaboration with the University of Canterbury. A number of these proposed modifications were incorporated in the town car, when alterations to the A40 Farina body structure were carried out.

As a town car, the requirement of passenger comfort is important, as is the need for a compact car able to accommodate four adults. This report examines an interconnected suspension used to achieve big-car ride comfort in a small/medium sized car, it examines the suspension geometry used to control pitch and roll, it covers the analysis of the rear suspension arms incorporating the drive motors and discusses the chassis layout based on an earlier report and methods of chassis construction. No attempt has been made to discuss the detail component design, although reductions of some 20 A1-size workshop drawings is included in Appendix 6.1.

CHAPTER 2

INTERCONNECTED SUSPENSION

2.1 HISTORY

Interconnected suspension - sometimes called Equalizing Suspension or Coupled Suspension - it is generally agreed, provides an improved ride compared to conventional suspension.

This agreement comes from both practical considerations of suspension design (Giles, 1960; Baker, 1973; Givens, 1975); and from mathematical analysis (Pevsner, 1957; Ellis, 1973; Bulman, 1976; Bastow, 1980).

The Citroen 2CV in 1949 was the first commercial application of interconnected suspension, followed by Packard in 1955. The Hydrolastic and Hydrogas cars from British Leyland are more recent examples.

A soft ride is traditionally synonymous with poor handling. Improve the handling and a harsh ride results. Interconnected suspension provides a means to reconcile this conflict, by reducing the national frequency of pitching.

Reduction of pitching is the main advantage of interconnected suspension. Pitching is particularly unpleasant to passengers. Lowering this frequency reduces the angular acceleration and improves the ride qualities. However, the closer the exciting frequency is to the natural frequency of the vehicle suspension the greater is the amplitude of oscillation. The use of interconnecting suspensions enables the natural frequency to be reduced to a value at which resonance is not normally encountered. In these circumstances the vibration due to road irregularities is purely vertical.

In an earlier report, Vandendungen (1976) a comparison was made between three cars to illustrate the effect of interconnected suspension. The cars approximate the Electric Town Car, and are identical to each other except for the suspension springing.

Car A has interconnected suspension, with the interconnection stiffness one-half that of the front or rear suspension. Car B has conventional suspension (interconnection stiffness zero) with the front and rear stiffness the same as car A. Car C has conventional suspension with the front and rear stiffness altered to produce the same bounce frequency as car A.

The results are as follows:

	Calculated Resonant Frequencies	
	Pitch	Bounce
Car A (Interconnected)	45 cpm	77 cpm
Car B (Conventional)	59 cpm	65 cpm
Car C (Conventional)	72 cpm	77 cpm

For the same bounce frequency, the pitch frequency for the interconnected car, might be expected to decrease by 27 cpm relative to the conventional car, depending on the interconnection stiffness used.

Similar results were achieved for the Austin Allegro with the Hydrogas Interconnected Suspension System, Baker (1973).

	Measured Resonant Frequencies	
	Pitch	Bounce
Hydrogas Interconnected Suspension	78 cpm	67 cpm
Typical continental small car with conventional suspension and good handling	104 cpm	85 cpm
	26 cpm difference.	

Packard cars in 1956 were using a bounce frequency of 54 cpm and a pitch frequency of 40 cpm, (Pevsner, 1957). With such low frequencies, the Packard required self-levelling, anti-dive, anti-squat and anti-roll bars. The Allegro with its high frequencies does not require these. As a comparison, the Rover 3500 and BMW 728, (Hartley, 1978) have the following nominal frequencies:

	Pitch	Bounce
Rover 3500	64 cpm	69 cpm
BMW 728	65 cpm	71 cpm

The BMW 728, for example, has anti-roll bars front and rear and anti-dive geometry incorporated at the front.

2.2 INTERCONNECTED SUSPENSION THEORY

2.2.1 General Principles

Figure 2.1 illustrates a simple interconnected suspension system. Between the body and the front and rear wheel assemblies on each side of the vehicle are springs C_3 and C_4 respectively. An interconnecting beam couples the front wheel assembly to the rear wheel assembly. Between each end of the beam and the respective wheel assemblies are additional springs C_1 and C_2 respectively. A pivot, distance d from the front wheel assembly and distance e from the rear secures the beam to the body on each side.

When the front wheel hits a bump, it rises and compresses C_1 and C_3. The reaction of C_1 is transmitted through the interconnecting beam to C_2 and hence to the rear wheel. Thus when either front wheel hits a bump, there is an increased reaction at both the front and rear wheels, causing that side of the vehicle to rise. If both front wheels hit a bump at the same time, the whole vehicle will rise. There is considerably less pitching compared to conventional suspension, and this is found to increase passenger ride comfort. Springs C_3 and C_4 are required to maintain pitch stability.

Where f_1 and f_2 are the front and rear wheel displacements relative to the body, and assuming linear springrates, then the front and rear wheel reactions P_1 and P_2 respectively can be expressed as follows:

$$\left.\begin{aligned} P_1 &= C_{11}f_1 + C_{12}f_2 \\ P_2 &= C_{21}f_1 + C_{22}f_2 \end{aligned}\right\} \qquad (2.2.1)$$

where C_{11}, C_{12}, C_{21} and C_{22} are relative stiffness values and are expressed in terms of C_1, C_2, C_3, C_4, d and e.

The stiffness C_{11} is the springrate at the front wheel due to the displacement of the front wheel and C_{21} is the springrate or stiffness at the rearwheel due to the displacement of the front wheels and so on. C_{11} and C_{22} are therefore basic stiffnesses, while C_{12} and C_{21} are interconnecting stiffnesses.

Where G_1 and G_2 are static front and rear wheel loads and f_{01} and f_{02} are static front and rear wheel displacements, equation (2.2.1) can be written as:

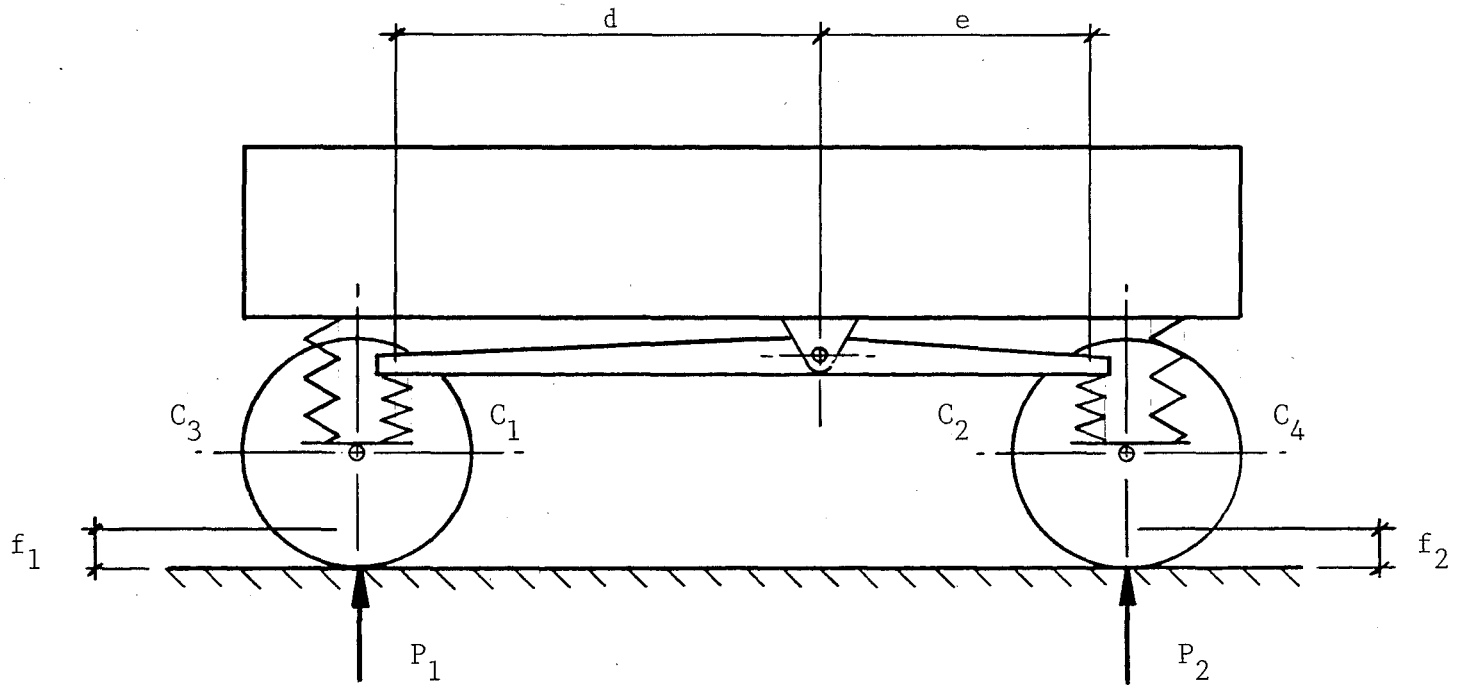


Figure 2.1. Simple Interconnected Suspension System.

$$G_1 = C_{11}f_{01} + C_{12}f_{02}$$

$$G_2 = C_{21}f_{01} + C_{22}f_{02}$$

Solving for f_{01} and f_{02}:

$$\left.\begin{aligned} f_{01} &= \frac{C_{22}\,G_1 - C_{12}\,G_2}{C_{11}\,C_{22} - C_{12}\,C_{21}} \\ f_{02} &= \frac{C_{11}\,G_2 - C_{21}\,G_1}{C_{11}\,C_{22} - C_{12}\,C_{21}} \end{aligned}\right\} \qquad (2.2.2)$$

2.2.2 Vibrations

Ignoring unsprung masses, the tyre stiffness, friction and viscous damping and with reference to figure 2.2.2 the equations of motion are;

$$\left.\begin{aligned} M\ddot{z} - P_{g1} - P_{g2} &= 0 \\ I_o\,\ddot{\psi} - P_{g1}a + P_{g2}b &= 0 \end{aligned}\right\} \qquad (2.2.3)$$

where M is the mass of the body, $I_o = MK^2$ the moment of inertia of the body relative to the lateral or transverse axis through the centre of gravity; K the radius of gyration; a and b the distances from the centre of gravity to the front and rear axles respectively, and P_{g1} and P_{g2} the dynamic load.

According to equation (2.2.1)

$$P_{g1} = C_{11}f_{g1} + C_{12}f_{g2}$$

$$P_{g2} = C_{21}f_{g1} + C_{22}f_{g2}$$

where f_{g1} and f_{g2} are the deflections of the front and rear springs under the dynamic load. Since

$$f_{g1} = q_1 - Z - a\,\psi$$

$$f_{g2} = q_2 - Z + b\,\psi$$

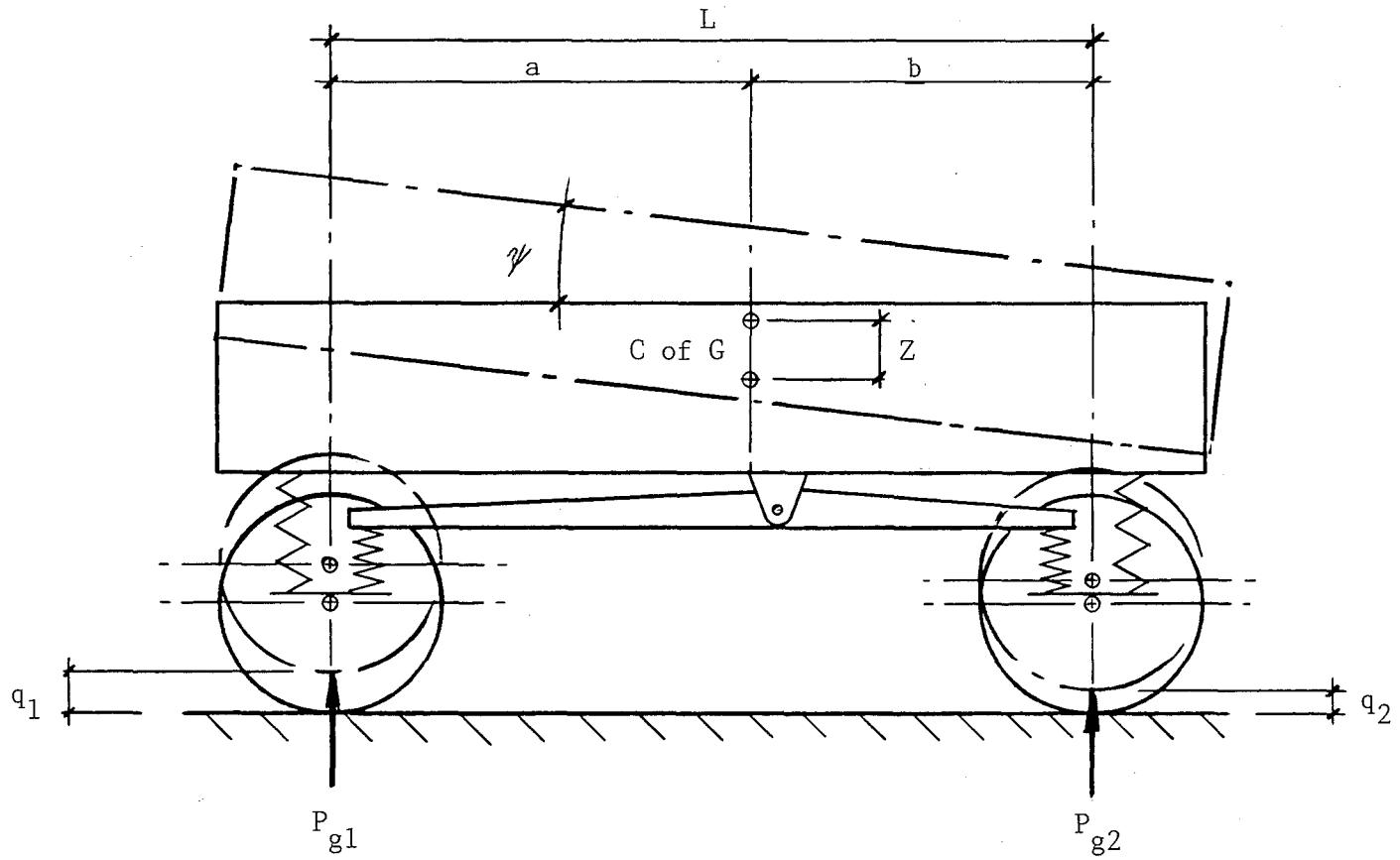


Figure 2.2.2. Bump Conditions.

Equation (2.2.3) becomes:

$$M\ddot{z} + (C_{11} + C_{22} + C_{12} + C_{21})z + [a(C_{11} + C_{21}) - b(C_{22} + C_{12})]\psi$$

$$= q_1(C_{11} + C_{21}) + q_2(C_{22} + C_{12})$$

$$I_o\ddot{\psi} + [a^2C_{11} + b^2C_{22} - ab(C_{12} + C_{21})]\psi + [a(C_{11} + C_{12}) - b(C_{22} + C_{21})]Z$$

$$= q_1(aC_{11} - bC_{21}) - q_2(bC_{22} - aC_{12}) \qquad (2.2.4)$$

The left-hand sides of equations (2.2.4) are the free vibrations of the system, while the right-hand sides are time dependent as q_1, q_2 are functions of time and represent the road irregularities.

The equations for free vibration are:

$$\left.\begin{aligned} \ddot{Z} + \omega_z^2 Z + \eta_{C1}\psi &= 0 \\ \ddot{\psi} + \omega_\alpha^2 \psi + \eta_{C2} Z &= 0 \end{aligned}\right\} \qquad (2.2.5)$$

where $\omega_z^2 = (C_{11} + C_{22} + C_{12} + C_{21})/M$ (2.2.6)

$$\omega_\alpha^2 = [a^2C_{11} + b^2C_{22} - ab(C_{12} + C_{21})]/MK^2 \qquad (2.2.7)$$

$$\eta_{C1} = [a(C_{11} + C_{21}) - b(C_{22} + C_{12})]/M \qquad (2.2.8)$$

$$\eta_{C2} = [a(C_{11} + C_{12}) - b(C_{22} + C_{21})]/MK^2 \qquad (2.2.9)$$

The magnitudes ω_z and ω_α are the partial frequencies of the system, that is, the frequencies which would apply if one of the coordinates Z or ψ is constant while the other is variable. If the coupling factors η_{C1} and η_{C2} are zero then the vertical and angular oscillations are independent, in which case the partial frequencies are the frequencies of the system.

Equations 2.2.5 can be shown to have the following solutions:

$$\psi = B_1 \text{ Sin } (\Omega_1 t + a_1) + B_2 \text{ Sin } (\Omega_2 t + a_2)$$

$$Z = B_1 \frac{\Omega_1^2 - \omega_\alpha^2}{\eta_{C2}} \text{ Sin } (\Omega_1 t + a_1) + B_2 \frac{\Omega_2^2 - \omega_\alpha^2}{C2} \text{ Sin } (\Omega_2 t + a_2) \qquad (2.2.10)$$

The national frequencies are given by

$$\Omega^2 = \frac{1}{2}\left(\omega_z^2 + \omega_\alpha^2 \pm \sqrt{(\omega_\alpha^2 - \omega_z^2)^2 + 4\eta_{C1}\,\eta_{C2}}\right) \qquad (2.2.11)$$

Equations (2.2.5), (2.2.10) and (2.2.11) apply equally well to conventional suspensions where there is no interconnection. Equations (2.2.6) and (2.2.7) show that the presence of the interconnection stiffness C_{12} and C_{21} between front and rear, increases the frequency of vertical vibration ω_z and decreases that of pitching ω_α. The reduction of pitching is of particular importance.

It can be seen from equation (2.2.7) that it is possible to reduce the pitching frequency to any required value. If $a^2C_{11} + b^2C_{22} = ab(C_{12} + C_{21})$, then the frequency would be zero and the system unstable owing to the absence of pitch stiffness in the suspension.

2.2.3 Pitch Angle Deflections

Application of a braking or acceleration load or a variation in the static load causes the body to assume a new position, both with regard to height and to angle.

If the load on the front wheels is increased by ΔG_1, then from equation (2.2.2) the static deflections will be increased thus:

$$\left.\begin{aligned} \Delta f_{01} &= \frac{C_{22}\,\Delta G_1}{C_{11}C_{22} - C_{12}C_{21}} \\ \Delta f_{02} &= \frac{-\,C_{21}\,\Delta G_1}{C_{11}C_{22} - C_{12}C_{21}} \end{aligned}\right\} \qquad (2.2.12)$$

The minus sign indicating that the deflection of the rear suspension is reduced as the load on the front wheels is increased. The corresponding angular displacement is:

$$\begin{aligned} \Delta\psi &= \frac{\Delta f_{02} - \Delta f_{01}}{L} \\ &= -\frac{C_{22} + C_{21}}{C_{11}C_{22} - C_{12}C_{21}} \quad \frac{\Delta G_1}{L} \end{aligned} \qquad (2.2.13)$$

where L is the vehicle wheelbase (= a + b).

Similarly for G_2 on the rear wheels:

$$\left.\begin{aligned} \Delta f_{01} &= \frac{-\ C_{12}\ \Delta G_2}{C_{11}C_{22} - C_{12}C_{21}} \\ \Delta f_{02} &= \frac{C_{11}\ \Delta G_2}{C_{11}C_{22} - C_{12}C_{21}} \end{aligned}\right\} \qquad (2.2.14)$$

$$\Delta\psi = \frac{f_{02} - f_{01}}{L}$$

$$= \frac{C_{11} + C_{12}}{C_{11}C_{22} - C_{12}C_{21}} \quad \frac{\Delta G_2}{L} \qquad (2.2.15)$$

These equations show that the additional deflections Δf_{01} and Δf_{02} and the angular displacement $\Delta\psi$ depend to a large extent on the stiffness of the interconnection system C_{12} and C_{21}; increasing the stiffness results in an increase in Δf_{01} and Δf_{02} and a considerable increase of the angle $\Delta\psi$. With a conventional suspension system in which $C_{12} = C_{21} = 0$, the additional deflection and angular displacement of the body due to a change in the front to rear weight distribution is less than with an interconnected suspension system.

For the three cars cited in section 2.1, the deflections and angular displacements due to the addition of a $\Delta G_2 = 750$ N load at the rear wheels with L = 2.4 m are as follows:

		Δf_{01}	Δf_{02}	$\Delta\psi$
Car A:	Interconnected	- 22 mm	49 mm	1 deg. 41 min.
Car B:	Conventional	0 mm	38 mm	0 deg. 54 min.
Car C:	Conventional	0 mm	26 mm	0 deg. 37 min.

Car B has the same front and rear spring rate C_{11} and C_{22} as car A, the angle of tilt is half that of car A. Car C has the same bounce frequency, but higher pitch frequency than car A, and the angle of tilt is a little over one-third that of the car with interconnected suspension, car A.

The sensitivity of interconnected suspension systems to the load distribution between front and rear wheels constitutes its main disadvantage and limits the possibility of increasing the coupling stiffness C_{12} and C_{21}. If these stiffness values are of a similar order to the stiffnesses C_{11} and C_{22}, then even a slight alteration of the load distribution between front and rear wheels leads to a substantial deflection of the suspension and pitching of the body. The example considered above has the coupling stiffnesses C_{12} and C_{21} as half the value of the stiffnesses C_{11} and C_{12}.

When a car is braked its front wheels are subjected to an additional vertical loading, while the loading on the rear wheels is reduced. For a constant braking effort T at the wheels, the change in vertical reactions at the front and rear wheels can be expressed approximately as follows:

$$\Delta P_1 = \frac{Th}{L}$$

$$\Delta P_2 = -\frac{Th}{L}$$

where h is the vehicle centre of gravity height and L the vehicle wheelbase.

Substituting P_1 and P_2 into equations (2.2.13) and (2.2.15) yields the angle of tilt due to braking effort T

$$\Delta\psi_T = -\frac{C_{11} + C_{22} + C_{12} + C_{21}}{C_{11}C_{22} - C_{12}C_{21}} \cdot \frac{Th}{L^2} \qquad (2.2.16)$$

Comparing the three cars again, at a deceleration rate of 0.6 g, assuming no anti-dive or anti-squat geometry, wheelbase L = 2.4 m, h = 0.6 m and M = 1000 kg:

		Body Pitch Angle Due to Braking	
Car A:	Interconnected	6 deg.	24 min.
Car B:	Conventional	3 deg.	21 min.
Car C:	Conventional	2 deg.	20 min.

As was the case with the uneven load distribution, the angle of pitch of the body with interconnected suspension is considerably greater than that of the cars with conventional suspension. In order to reduce the angle to an acceptable level during braking and accelerating, anti-drive and anti-squat geometry needs to be incorporated into the suspension design.

2.3 APPLICATION OF AN INTERCONNECTED SUSPENSION SYSTEM

2.3.1 Previous Work

In an earlier report, Vandendungen (1976), an interconnected suspension system was proposed. This is shown in figure 2.3.1 which illustrates the lefthand side of a vehicle. The front, rear and auxiliary torsion cars are necessarily less than one-half of the vehicle width.

When the front wheel hits a bump, this causes the front torsion car to wind up (1) and apply additional tension in the cable (3). This additional tension causes the rear torsion car to wind up (4) and apply additional load at the rear wheels (2). Hitting a bump with one front wheel causes an increased reaction at both the front and rear wheels causing that side of the car to rise. Pitching is thus reduced.

If the pitch arms which connect the torsion cars via the cable are inclined towards each other as in figures 2.3.2a and 2.3.2b, the springrates become non-linear and some pitch stability is induced. In case this effect was insufficient, an appropriate, auxiliary torsion car, fixed to the chassis at the inner end and acting like spring C_4 in figure 2.2.1, would increase the pitch stiffness to the desired level. It might also be used to adjust the static pitch angle by rotation of its anchored end.

The proposed suspension was analysed in 1976 and the springrates deduced, using the nomenclature from figures 2.3.2 and 3.

$$C_{11} = \frac{C_1 C_2}{\ell_1^2 \, (C_2 + C_1/R)} \tag{2.3.1}$$

$$C_{22} = \frac{(C_1 C_2 + C_3 C_1 + C_2 C_3 R)}{\ell_2^2 \, (C_1 + C_2 R)} \tag{2.3.2}$$

$$C_{12} = \frac{\ell_3}{\ell_4} \, \frac{C_1 C_2}{\ell_1 \ell_2 \, (C_1 + C_2 R)} \tag{2.3.3}$$

$$C_{21} = \frac{\ell_4}{\ell_3} \, \frac{C_1 C_2}{\ell_1 \ell_2 \, (C_2 + C_1/R)} \tag{2.3.4}$$

A computer program was written later, as part of this M.E. project (see section 2.4), to investigate the suspension system. This showed that in practice insufficient pitch stability could be achieved from the

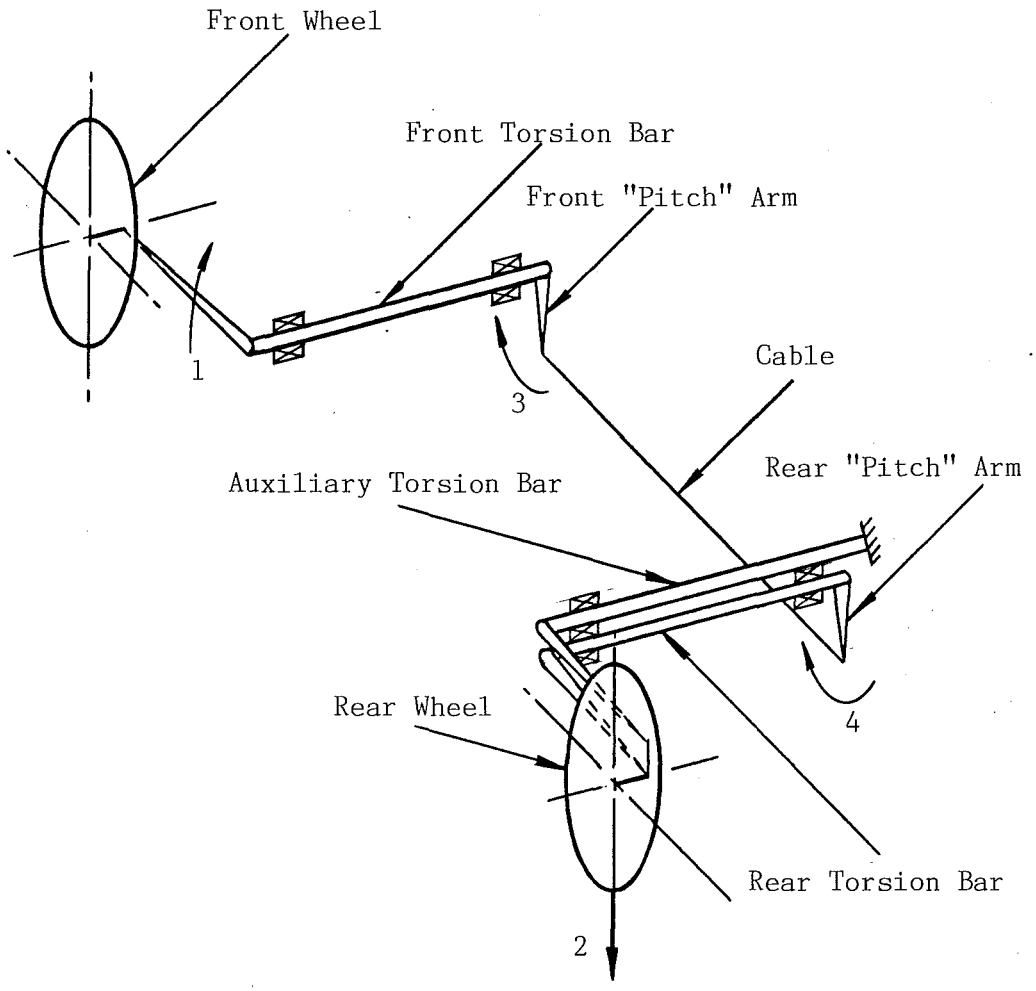


Figure 2.3.1. Original Proposed Design of Interconnected Suspension

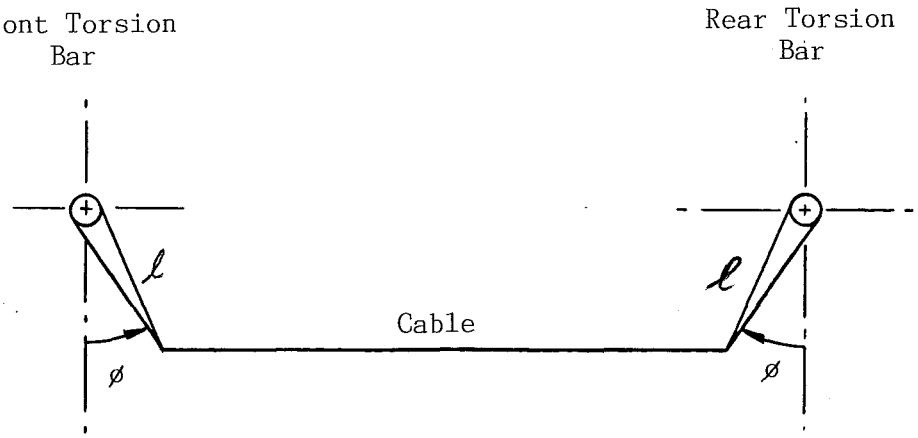


Figure 2.3.2.a. Pitch Arms in Undeflected Position.

$\ell_3 = \ell \cos(\phi - \Theta_3)$ $\qquad$ $\ell_4 = \ell \cos(\phi + \Theta_4)$

Figure 2.3.2.b. Pitch Arms in Deflected Position.

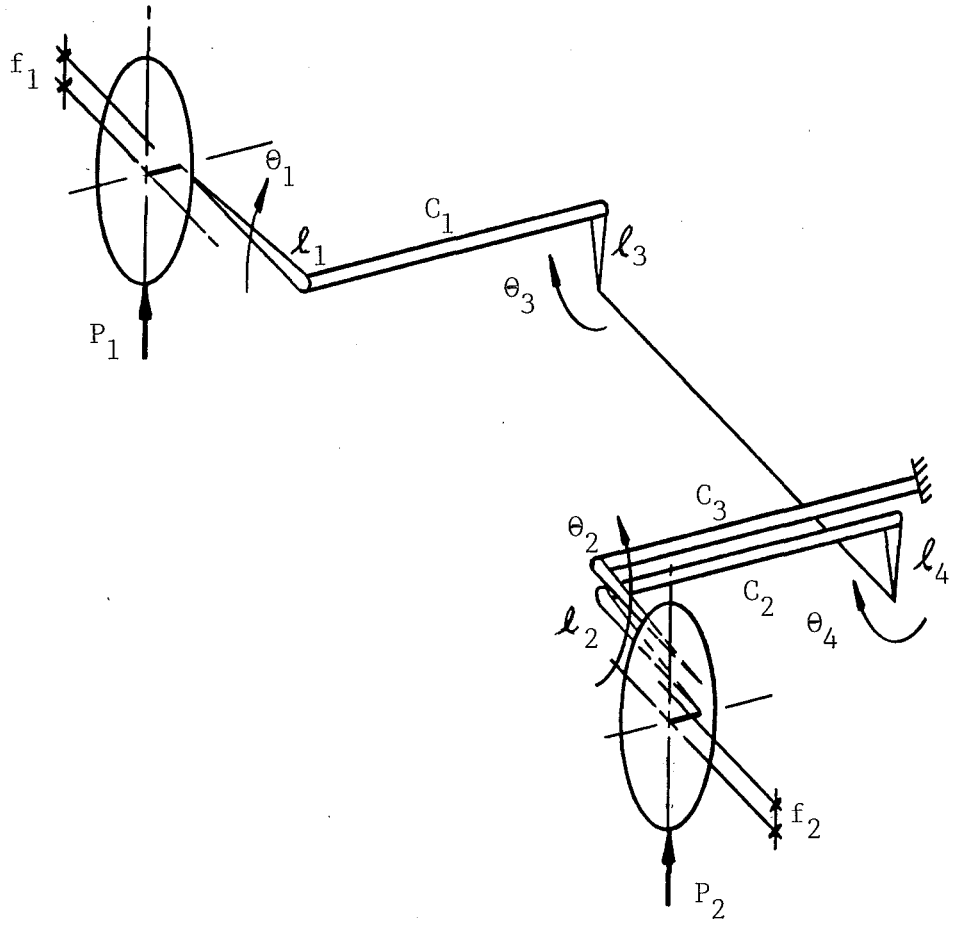


Figure 2.3.3. Nomenclature.

f_1 Front wheel displacement relative to body.
f_2 Rear wheel displacement relative to body.
P_1 Reaction on front wheels.
P_2 Reaction on rear wheels.
ℓ_1 Effective length of front leading arm.
ℓ_2 Effective length of rear trailing arm.
ℓ_3 Effective length of front pitch arm.
ℓ_4 Effective length of rear pitch arm.
C_1 Stiffness of front torsion bar.
C_2 Stiffness of rear torsion bar.
C_3 Stiffness of auxiliary torsion bar.
θ_1 Angle deviation from static for arm ℓ_1.
θ_2 Angle deviation from static for arm ℓ_2.
θ_3 Angle deviation from static for arm ℓ_3.
θ_4 Angle deviation from static for arm ℓ_4.
R Ratio $\ell_3\theta_4/\ell_4\theta_3$.

inclined pitch arms alone, as the angular deflection was too small to gain the required effect. The auxiliary torsion bar would therefore be required, but it could better be used to control the motion of the interconnecting cable.

2.3.2 Suspension as Built

The suspension as built utilizes three torsion bars on each side of the vehicle. The auxiliary torsion bar has one end connected to the rear pitch arm, rather than the rear suspension arm, and the other end fixed to the chassis as in figure 2.3.4. Using the nomenclature as previously, θ_4 is common to both the rear torsion bar C_2 and the auxiliary torsion bar C_3.

Assume that the front wheels have been displaced by an amount f_1, relative to the body, figure 2.3.4. Torsion bar C_1 will twist by an angle $\theta_1 - \theta_3$ and the tension T in the cable will increase. Torsion bar C_2 will twist by an angle $\theta_4 - \theta_2$ and torsion bar C_3 will twist by an angle θ_4. The reaction P_1 on the front wheels will increase, as will the reaction P_2 on the rear wheels and the car will rise vertically. The reaction of torsion bar C_3 on the vehicle body maintains the pitch stability.

From equations (2.2.1), the vehicle springrates are derived as follows:

(a) Fix the rear wheels relative to the body, so that $f_2 = 0$, and equations (2.2.1) become,

$$P_1 = C_{11} f_1$$

$$P_2 = C_{21} f_1$$

thus C_{11} and C_{21} are derived.

(b) Fix the front wheels relative to the body so that $f_1 = 0$ and equations (2.2.1) become,

$$P_2 = C_{22} f_2$$

$$P_1 = C_{12} f_2$$

thus C_{22} and C_{21} are derived.

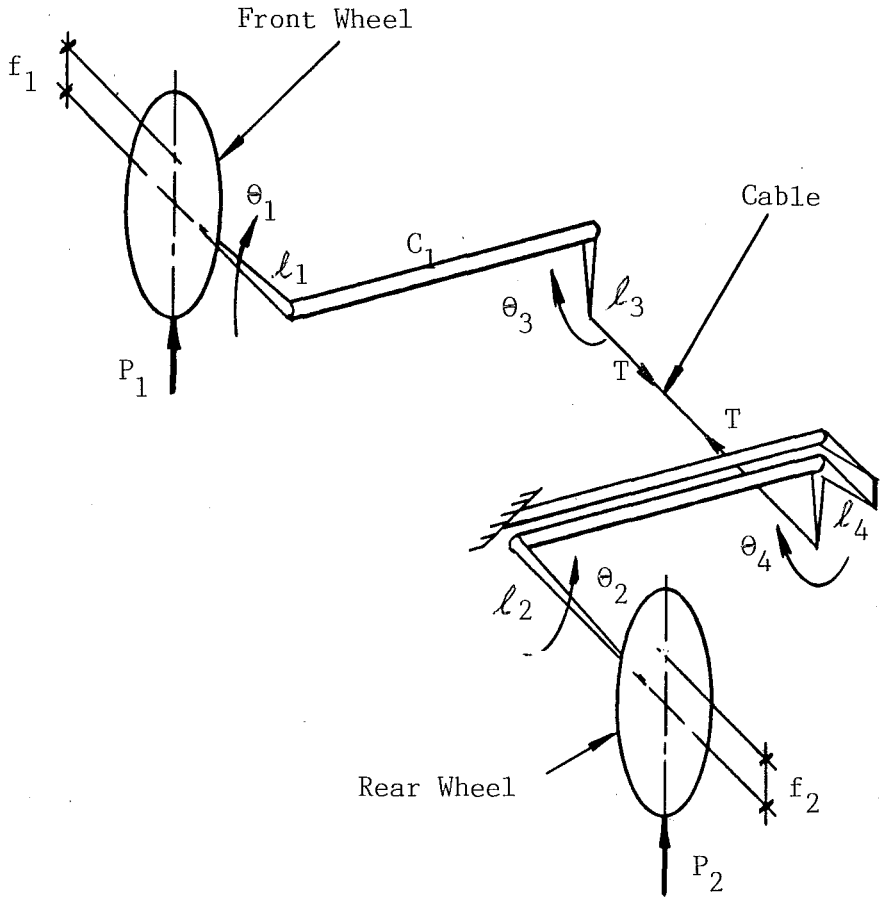
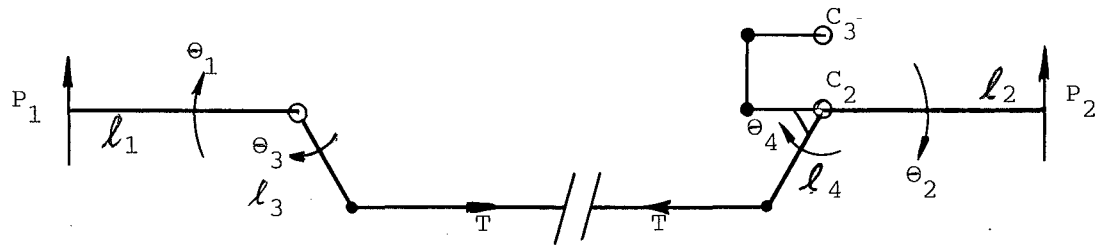


Figure 2.3.4. Torsion Bar Interconnected Suspension.

(a) Front wheel displacement.



$\Theta_1 > \Theta_3 \qquad \Theta_2 = 0$

Take moments about torsion bars C_1 and C_2.

(1) $P_1 \ell_1 = T\ \ell_3 \Rightarrow T = P_1 \ell_1 / \ell_3$

(2) $P_2 \ell_2 = T\ \ell_4 - C_3 \theta_4$

Torsion bar moments.

(3) $P_1 \ell_1 = C_1(\theta_1 - \theta_3) = (C_1 f_1/\ell_1 - C_1 \theta_3)$

(4) $P_2 \ell_2 = C_2 \theta_4$.

Substitute (2) into (4)

$$C_2 \theta_4 = T\ell_4 - C_3 \theta_4$$

Substitute (1)

$$C_2 \theta_4 = \frac{P_1 \ell_1 \ell_4}{\ell_3} - C_3 \theta_4$$

$$\Rightarrow \theta_4 (C_2 + C_3) = P_1 \ell_1\ \ell_4/\ell_3.$$

(5) $$\theta_4 = \frac{P_1 \ell_1}{(C_2 + C_3)}\ \frac{\ell_4}{\ell_3}$$

Substitute (5) into (3)

$$P_1 \ell_1 = \frac{C_1 f_1}{\ell_1} - \frac{C_1 \theta_3}{\theta_4} \cdot \theta_4 = \frac{C_1 f_1}{\ell_1} - C_1 \frac{\theta_3}{\theta_4}\ \frac{P_1 \ell_1}{(C_2 + C_3)}\ \frac{\ell_4}{\ell_3}$$

and if $$R = \frac{\ell_3}{\ell_4}\ \frac{\theta_4}{\theta_3}$$

$$P_1 \ell_1 \left(1 + \frac{C_1}{(C_2+C_3)} \cdot \frac{1}{R}\right) = \frac{C_1 f_1}{\ell_1}$$

$$(6) \quad P_1 = \frac{C_1(C_2+C_3)}{\ell_1^2\,(C_2+C_3+C_1/R)}\, f_1$$

so that $$C_1 = \frac{C_1(C_2+C_3)}{\ell_1^2(C_2+C_3+C_1/R)} \qquad (2.3.5)$$

For C_{21}, substitute (1) and (5) into

$$P_2 \ell_2 = P_1 \ell_1 \frac{\ell_4}{\ell_3} - \frac{C_3}{C_2+C_3} \frac{\ell_4}{\ell_3} P_1 \ell_1$$

$$(7) \quad P_2 = \frac{P_1 \ell_1}{\ell_2} \frac{\ell_4}{\ell_3} \left(1 - \frac{C_3}{(C_2+C_3)}\right)$$

substitute (6) into (7)

$$P_2 = \frac{\ell_4}{\ell_3} \frac{C_1 C_2}{\ell_1 \ell_2 (C_2+C_3+C_1/R)}\, f_1$$

so that $$C_{21} = \frac{\ell_4}{\ell_3} \frac{C_1 C_2}{\ell_1 \ell_2 (C_2+C_3+C_1/R)} \qquad (2.3.6)$$

(b) Rear wheel displacement.

$$\theta_2 > \theta_4 , \quad \theta_1 = 0.$$

Take moments about torsion bars C_1 and C_2

$$(1) \quad P_1 \ell_1 = T \ell_3$$

$$(2) \quad P_2 \ell_2 = T \ell_4 + C_3 \theta_4 \Rightarrow T = \frac{P_2 \ell_2}{\ell_4} - \frac{C_3 \theta_4}{\ell_4}$$

Torsion bar moments

$$(3) \quad P_1 \ell_1 = C_1 \theta_3$$

$$(4) \quad P_2 \ell_2 = C_2(\theta_2 - \theta_4) = C_2\, f_2/\ell_2 - C_2 \theta_4$$

substitute (1) into (3)

$$C_1\theta_3 = T\ell_3$$

substitute (2)

$$C_1\theta_3 = P_2\ell_2 \frac{\ell_3}{\ell_4} - C_3\theta_4 \frac{\ell_3}{\ell_4}$$

$$\Rightarrow \quad C_1 \frac{\theta_3}{\theta_4} \cdot \theta_4 = P_2\ell_2 \frac{\ell_3}{\ell_4} - C_3\theta_4 \frac{\ell_3}{\ell_4}$$

$$(5) \qquad \theta_4 = \frac{P_2\ell_2 R}{(C_1+C_3R)}$$

substitute (5) into (4)

$$P_2\ell_2 = \frac{C_2 f_2}{\ell_2} - \frac{C_2 P_2 \ell_2 R}{(C_1+C_3 R)}$$

$$(6) \qquad P_2 = \frac{C_2(C_1+C_3 R)}{\ell_2^2(C_1+C_2R+C_3R)} f_2$$

so that $$C_{22} = \frac{C_2(C_1+C_3R)}{\ell_2^2(C_1+C_2R + C_3R)} \qquad (2.3.7)$$

substitute (2) and (5) into (1)

$$P_1\ell_1 = P_2\ell_2 \frac{\ell_3}{\ell_4} - P_2\ell_2 \frac{\ell_3}{\ell_4} \frac{C_3R}{(C_1+C_3R)}$$

$$= P_2\ell_2 \frac{\ell_3}{\ell_4} \left(1 - \frac{C_3R}{(C_1+C_3R)}\right)$$

$$(7) \qquad P_1 = P_2 \frac{\ell_2}{\ell_1} \frac{\ell_3}{\ell_4} \left(1 - \frac{C_3R}{(C_1+C_3R)}\right)$$

substitute (6) into (7)

$$P_1 = \frac{\ell_3}{\ell_4} \frac{C_2(C_1+C_3R)}{\ell_1\ell_2(C_1+C_2R+C_3R)} \cdot \frac{C_1}{(C_1+C_3R)} f_2$$

$$(8) \qquad P_1 = \frac{\ell_3}{\ell_4} \; \frac{C_1C_2}{\ell_1\ell_2(C_1+C_2R+C_3R)} \; f_2$$

so that $$C_{12} = \frac{\ell_3}{\ell_4} \; \frac{C_1C_2}{\ell_1\ell_2(C_1+C_2R+C_3R)} \qquad (2.3.8)$$

Assuming linear springrates, the suspension is unstable if $a^2C_{11} + b_2C_{22} = ab(C_{12}+C_{21})$. (See section 2.2.2). In the case where a = b, then $C_{11} + C_{22} = C_{12} + C_{21}$. Substituting in C_{11}, C_{22}, C_{21} and C_{21} we have

$$\frac{C_1(C_2+C_3)}{\ell_1^2(C_2+C_3+C_1/R)} + \frac{C_2(C_1+C_3R)}{\ell_2^2(C_1+C_2R+C_3R)} = \frac{\ell_4}{\ell_3} \; \frac{C_1C_2}{\ell_1\ell_2(C_2+C_3+C_1/R)} + \frac{\ell_3}{\ell_4} \; \frac{C_1C_2}{\ell_1\ell_2(C_1+C_2R+C_3R)}$$

In this case where a = b, then $P_1 = P_2$ and $P_1 = \frac{T\ell_3}{\ell_1} = P_2 = \frac{T\ell_4}{\ell_2}$

Hence $$\frac{\ell_3}{\ell_4} = \frac{\ell_1}{\ell_2}$$

Substituting $\ell_3/\ell_4 = \ell_1/\ell_2$ and manipulating R:

$$\frac{RC_1(C_2+C_3)}{\ell_1^2(C_2R+C_3R+C_1)} + \frac{C_2(C_1+C_3R)}{\ell_2^2(C_1+C_2R+C_3R)} = \frac{C_1C_2R}{\ell_1^2(C_1+C_2R+C_3R)} + \frac{C_1C_2}{\ell_2^2(C_1+C_2R+C_3R)} \; .$$

Multiply through by $(C_1+C_2R+C_3R)\ \ell_1^2\ \ell_2^2$ so that $\ell_2^2\ R_1C_1(C_2+C_3) + \ell_1^2\ C_2(C_1+C_3R) =$

$\ell_2^2C_1C_2R+\ell_1^2\ C_1C_2$ which leads to $\ell_2^2(C_1C_2R-C_1C_2R-C_1C_3R) + \ell_1^2(C_1C_2-C_1C_2-C_2C_3R) = 0$

and hence $C_3(C_1\ell_2^2 + C_2\ell_1^2) = 0$ for instability. Thus the system is unstable if $C_3 = 0$, that is, if the third or auxiliary torsion bar is not present.

2.3.3 Comparison of Systems

The springrates of the two proposed torsion bar systems are summarized as follows:

Auxiliary torsion bar position:	on suspension arm	on pitch arm
C_{11}	$\frac{C_1C_2}{\ell_1^2(C_2+C_1/R)}$	$\frac{C_1(C_2+C_3)}{\ell_1^2(C_2+C_3+C_1/R)}$
C_{22}	$\frac{(C_1C_2+C_1C_3+C_2C_3R)}{\ell_2^2(C_1+C_2R)}$	$\frac{C_2(C_1+C_3R)}{\ell_2^2(C_1+C_2R+C_3R)}$
C_{12}	$\frac{\ell_3}{\ell_4}\frac{C_1C_2}{\ell_1\ell_2(C_1+C_2R)}$	$\frac{\ell_3}{\ell_4}\frac{C_1C_4}{\ell_1\ell_2(C_1+C_2R+C_3R)}$
C_{21}	$\frac{\ell_4}{\ell_3}\frac{C_1C_2}{\ell_1\ell_2(C_2+C_1/R)}$	$\frac{\ell_4}{\ell_3}\frac{C_1C_2}{\ell_1\ell_2(C_2+C_3+C_1/R)}$

With the auxiliary torsion bar C_3 connected to the rear suspension arm, C_3 affects and increases only C_{22}. With the auxiliary torsion bar C_3 connected to the pitch arms, C_3 affects all the springrates, increasing the basic stiffnesses C_{11} and C_{22} and decreasing the interconnecting stiffnesses C_{12} and C_{21}. This system is versatile, enabling C_{11}, C_{22} and the coupling stiffnesses as a pair, to be altered separately, by using different combinations of C_1, C_2 and C_3.

Consider a numerical example for a vehicle which approximates the Electric Town Car, with M = 1050 kg, K = 0.49 L, a = 0.449 L, b = 0.551 L, ℓ_1 = 0.45, ℓ_2 = 0.51 and L = 2.23. From these K^2/ab = 0.97 which is possibly a little too large. For ease of calculation assume R=1 and ℓ_3/ℓ_4 = 1.

	C_3 on pitch arm	C_3 on rear arm case 1	C_3 on rear arm case 2
C_1	3638	3638	1500
C_2	3638	3638	1500
C_3	3181	3181	3181
C_{11}	23430	17966	7407
C_{22}	18242	38446	30227
C_{21}	11030	15852	6536
C_{12}	11030	15852	6536

	C_3 on pitch arm	C_3 on rear arm case 1	C_3 on rear arm case 2
w_z^2	60.698	83.920	48.292
w_α^2	19.046	29.552	29.494
η_{c1}	-1.410	-31.314	-29.737
η_{c2}	-1.180	-26.212	-24.892
Bounce Hz	1.24	1.56	1.31
Pitch Hz	0.694	0.661	0.506
Bounce cpm	74	94	79
Pitch cpm	42	40	30
Body tilt 750 N rear luggage	2.2^o	1.5^o	1.5^o
Body tilt 0.6 g braking	-7.3^o	-7.0^o	-9.8^o

With the torsion car C_3 attached to the rear suspension arm, the difference between bounce and pitch frequencies was limited to approximately 50 cpm and the ratio of bounce to pitch did not go below 2.3 for all the different spring combinations of C_1, C_2 and C_3 tried. This layout proved to be limited.

2.4 SUSPENSION ANALYSIS ON A DIGITAL COMPUTER

2.4.1 The Interconnected Suspension Program "Intersusp"

The object of writing this program was to establish the pitch characteristics of the inclined arms which interconnect the main tension bars via cables and more importantly to calculate the position and condition of each suspension component for every front and rear, bump and rebound permutation and hence the vehicle's suspension characteristics and vibration responses at each permutation.

As discussed in section 2.3.1 the inclined pitch arms alone do not contribute sufficient pitch stability and a third torsion bar is required. These so-called pitch arms are retained inclined towards each other (rather than parallel) as they produce a non-linear rising-springrate effect.

The program analyses the suspension proposed in section 2.3.2 and uses equations (2.3.5) to (2.3.8) inclusive for C_{11}, C_{21}, C_{22} and C_{12}. For every permutation of bounce and rebound of F_1 and F_2 analysed, the instantaneous values of θ_1, θ_2, θ_3, θ_4, ℓ_1, ℓ_2, ℓ_3, ℓ_4 and R are calculated. The angles θ_3 and θ_4 and lengths ℓ_3 and ℓ_4 are not independent and a Modified Newton-Raphson Iteration Method is employed. Using the analysis in section 2.2.2 the corresponding partial bounce and pitch frequencies and coupling coefficients are calculated and hence the bounce and pitch frequencies.

In addition to the dynamic wheel loadings P_1 and P_2, the ratio of dynamic to static wheel loadings are calculated. To aid the mechanical design, at every permutation of F_1 and F_2, the cable tension and the windup angles and stresses for all three torsion bars is also calculated.

The computer printout lists the vehicle data, some static condition suspension properties and finally the bump and rebound analysis. A guide on how to input the data cards is given in the booklet "Interconnected Suspension Program, Data Input and Answer Code Handbook", Appendix 2.4.1.

With reference to the sample printout, the "Program Control Integers" comprise the first data card. The value of the integers control a subroutine DATALIST which, in this example sets up the program to expect circular rather than rectangular front and rear torsion bars, no Load-Brake-Acceleration Data, a rectangular auxiliary torsion bar and so on. Separate cards are then read for the data on the front torsion bar, rear torsion bar, arm lengths, initial angles, vehicle geometry and auxiliary torsion bar. This data is echoed on the printout.

The springrate and stressrate for the three torsion bars is calculated, SAE (1947) and printed.

A subroutine ARMLTH is called to calculate the required effective armlength of the rear pitch arm L4, based on the vehicle geometry. Armlengths AL3 and AL4 are the required physical centre to centre lengths of the front and rear pitch arms, while AL6 is the corresponding centre to centre length of the cables. The locked pitch angles refers to the condition when the four-car-linkage system comprising front and rear pitch arms and the cable becomes locked, forming a triangle such that the cable and one arm are in line.

The static deflections front and rear are calculated in subroutine STDEFL, assuming that the inboard ends of the front and rear torsion bars are fixed. Equations (2.2.2) were not used, as these assume C_{11}, C_{22},

```
PROGRAM CONTROL INTEGERS
ITF =    1  FRONT TORSION BAR (1=CIRCULAR,2=RECTANGULAR)
ITR =    1  REAR TORSION BAR (1=CIRCULAR,2=RECTANGULAR)
IAL =    1  ARM LENGTH DATA
IAD =    1  INITIAL ANGLE DATA
IVG =    1  VEHICLE GEOMETRY DATA
LBA =    0  LOAD-BRAKE-ACCELERATION DATA
ITA =    2  AUX TORSION BAR (1=CIRCULAR,2=RECTANGULAR)
   0

FRONT TORSION BAR * * CIRCULAR * * ITF =    1
      G           L           D
.78453E+11    0.4960      0.0220     0.0000     0.0000     0.0000     0.0000     0.0000

REAR TORSION BAR * * CIRCULAR * * ITR =    1
      G           L           D
.78453E+11    0.4960      0.0220     0.0000     0.0000     0.0000     0.0000     0.0000

ARM LENGTH DATA * * IAL =    1
     AL1         AL2         AL3         AL4         AL5         AL6          L3
   0.4500      0.5090      0.0000      0.0000      1.3500      0.0000      0.0700      0.0000

INITIAL ANGLE DATA * * IAD =    1
      A1          A2          A3          A4          A5          A6
   0.0000      0.0196      1.0472      1.0472      0.0000      0.0000      0.0000      0.0000

VEHICLE GEOMETRY DATA * * IVG =    1
      WB           A           B           H           M           K
   2.2300      0.4490      0.5510      0.2200   1049.6000      0.4900      0.0000      0.0000

AUXILIARY TORSION BAR * * RECTANGULAR * * ITA =    2
      G           L           A           B          N2          N3
.78453E+11    0.4560      0.0190      0.0190      0.2100      0.1400      0.0000      0.0000

EACH FRONT TORSION BAR * TORSION RATE C1 =      3.638 KNM/RAD
                       * STRESS-RATE     =   1739.885 MPA/RAD

EACH REAR TORSION BAR  * TORSION RATE C2 =      3.638 KNM/RAD
                       * STRESS RATE     =   1739.885 MPA/RAD

EACH AUXILIARY TOR BAR * TORSION RATE C3 =      3.181 KNM/RAD
                       * STRESS-RATE     =   2208.307 MPA/RAD

SUBROUTINE * ARMLTH * HAS BEEN CALLED

EFFECTIVE LENGTH L4 =     0.0646
ARM LENGTHS * AL3 =     0.0806  AND AL4 =     0.0748 METRES
CABLE LENGTH AL6 =     1.2723 AND INITIAL CABLE ANGLE A6 =    -0.0040 RADIANS

FRONT BUMP * LOCKED PITCH ANGLES ARE * FRONT =     1.5048 AND REAR =     0.0598 RADIANS
REAR BUMP * LOCKED PITCH ANGLES ARE * FRONT =     0.0553 AND REAR =     1.5824 RADIANS

SUBROUTINE * SIDEFI * HAS BEEN CALLED

*       F1          F2          P1          P2        THT1        THT2        THT3        THT4       P/G1       P/G2      RATIO           T         ITS
    0.0000      0.0000      2.8357      2.3108      0.0000      0.0000      0.0000      0.0000      1.0000      1.0000      1.0832     18.2297           0
INITIAL THT1 =    -0.3508 * INITIAL THT2 =     0.3239
STATIC DEFLECTIONS * FRONT =    -0.1546 * REAR =    -0.1617 M

SUBROUTINE * BUMP * HAS BEEN CALLED
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SUBROUTINE * BUMP * HAS BEEN CALLED

*        F1          F2          P1          P2          THT1        THT2        THT3        THT4        P/G1        P/G2        RATIO          T          ITS
  -0.1000    -0.1000     1.0784      0.8942     -0.2241      0.1978     -0.0034     -0.0036      0.3803      0.3869      1.1658     6.7734            8
CORRES WHEEL RATES ARE * C11    12.9642 * C22     9.3031 * C12     5.6386 * C21     5.6507 KN/M
CORRES FREQS ARE * PITCH     0.7291 * BOUNCE     1.2727 HZ        PFB =    1.2727 PFP =     0.7292 HZ        C COEFFS =     0.5322     0.4026
CORRES ANGLES * PHI1 =      0.1301 * PHI2 =    -0.1224 RAD AND STRESS * S1 =   226.3136 * S2 =  -212.9384 * S3 =    -8.0358 MPA

*        F1          F2          P1          P2          THT1        THT2        THT3        THT4        P/G1        P/G2        RATIO          T          ITS
  -0.1000    -0.0500     1.3858      1.3133     -0.2241      0.0983     -0.0404     -0.0427      0.4887      0.5683      1.0872     8.9135            6
CORRES WHEEL RATES ARE * C11    12.6763 * C22     9.1100 * C12     5.5490 * C21     5.6918 KN/M
CORRES FREQS ARE * PITCH     0.7146 * BOUNCE     1.2626 HZ        PFB =    1.2626 PFP =     0.7146 HZ        C COEFFS =     0.7232     0.0974
CORRES ANGLES * PHI1 =      0.1972 * PHI2 =    -0.1828 RAD AND STRESS * S1 =   290.8228 * S2 =  -318.0702 * S3 =   -94.3138 MPA

*        F1          F2          P1          P2          THT1        THT2        THT3        THT4        P/G1        P/G2        RATIO          T          ITS
  -0.1000     0.0000     1.6902      1.7424     -0.2241      0.0000     -0.0772     -0.0797      0.5960      0.7540      1.0160    11.1538            6
CORRES WHEEL RATES ARE * C11    12.3904 * C22     9.1003 * C12     5.5100 * C21     5.7814 KN/M
CORRES FREQS ARE * PITCH     0.7044 * BOUNCE     1.2578 HZ        PFB =    1.2579 PFP =     0.7043 HZ        C COEFFS =     0.4625    -0.5784
CORRES ANGLES * PHI1 =      0.2039 * PHI2 =    -0.2442 RAD AND STRESS * S1 =   354.7019 * S2 =  -424.8568 * S3 =  -175.9211 MPA

*        F1          F2          P1          P2          THT1        THT2        THT3        THT4        P/G1        P/G2        RATIO          T          ITS
  -0.1000     0.0500     1.9938      2.1975     -0.2241     -0.0981     -0.1138     -0.1149      0.7031      0.9510      0.9503    13.5278            8
CORRES WHEEL RATES ARE * C11    12.1025 * C22     9.2676 * C12     5.5183 * C21     5.9223 KN/M
CORRES FREQS ARE * PITCH     0.6978 * BOUNCE     1.2585 HZ        PFB =    1.2584 PFP =     0.6979 HZ        C COEFFS =    -0.2288    -1.6296
CORRES ANGLES * PHI1 =      0.2405 * PHI2 =    -0.3071 RAD AND STRESS * S1 =   418.4131 * S2 =  -534.2985 * S3 =  -253.7389 MPA

*        F1          F2          P1          P2          THT1        THT2        THT3        THT4        P/G1        P/G2        RATIO          T          ITS
  -0.1000     0.1000     2.2987      2.6973     -0.2241     -0.1970     -0.1505     -0.1487      0.8106      1.1673      0.8888    16.0748            8
CORRES WHEEL RATES ARE * C11    11.8089 * C22     9.6265 * C12     5.5743 * C21     6.1217 KN/M
CORRES FREQS ARE * PITCH     0.6942 * BOUNCE     1.2655 HZ        PFB =    1.2646 PFP =     0.6960 HZ        C COEFFS =    -1.3802    -3.1040
CORRES ANGLES * PHI1 =      0.2673 * PHI2 =    -0.3722 RAD AND STRESS * S1 =   462.3903 * S2 =  -647.5152 * S3 =  -328.4595 MPA

*        F1          F2          P1          P2          THT1        THT2        THT3        THT4        P/G1        P/G2        RATIO          T          ITS
  -0.0500    -0.1000     1.6744      1.1925     -0.1113      0.1978      0.0336      0.0372      0.5905      0.5161      1.2526    10.4898            8
CORRES WHEEL RATES ARE * C11    12.7555 * C22     9.1836 * C12     5.5213 * C21     5.4020 KN/M
CORRES FREQS ARE * PITCH     0.7308 * BOUNCE     1.2595 HZ        PFB =    1.2594 PFP =     0.7309 HZ        C COEFFS =     0.2137     0.6037
CORRES ANGLES * PHI1 =      0.2059 * PHI2 =    -0.1632 RAD AND STRESS * S1 =   358.1680 * S2 =  -283.9918 * S3 =    82.1470 MPA

*        F1          F2          P1          P2          THT1        THT2        THT3        THT4        P/G1        P/G2        RATIO          T          ITS
  -0.0500    -0.0500     1.9606      1.6078     -0.1113      0.0983     -0.0016     -0.0017      0.6914      0.6958      1.1698    12.5379            6
CORRES WHEEL RATES ARE * C11    12.4914 * C22     8.9891 * C12     5.4391 * C21     5.4445 KN/M
CORRES FREQS ARE * PITCH     0.7166 * BOUNCE     1.2499 HZ        PFB =    1.2498 PFP =     0.7166 HZ        C COEFFS =     0.4390     0.3482
CORRES ANGLES * PHI1 =      0.2410 * PHI2 =    -0.2238 RAD AND STRESS * S1 =   419.3796 * S2 =  -389.4050 * S3 =    -3.7739 MPA

*        F1          F2          P1          P2          THT1        THT2        THT3        THT4        P/G1        P/G2        RATIO          T          ITS
  -0.0500     0.0000     2.2444      2.0353     -0.1113      0.0000     -0.0365     -0.0386      0.7915      0.8808      1.0953    14.6753            4
CORRES WHEEL RATES ARE * C11    12.2307 * C22     8.9751 * C12     5.4070 * C21     5.5325 KN/M
CORRES FREQS ARE * PITCH     0.7067 * BOUNCE     1.2456 HZ        PFB =    1.2456 PFP =     0.7067 HZ        C COEFFS =     0.2171    -0.2647
CORRES ANGLES * PHI1 =      0.2759 * PHI2 =    -0.2852 RAD AND STRESS * S1 =   480.0846 * S2 =  -496.2892 * S3 =   -85.2572 MPA

*        F1          F2          P1          P2          THT1        THT2        THT3        THT4        P/G1        P/G2        RATIO          T          ITS
  -0.0500     0.0500     2.5279      2.4910     -0.1113     -0.0981     -0.0713     -0.0739      0.8914      1.0780      1.0269    16.9317            6
CORRES WHEEL RATES ARE * C11    11.9696 * C22     9.1351 * C12     5.4220 * C21     5.6687 KN/M
CORRES FREQS ARE * PITCH     0.7005 * BOUNCE     1.2467 HZ        PFB =    1.2466 PFP =     0.7007 HZ        C COEFFS =    -0.4307    -1.2384
CORRES ANGLES * PHI1 =      0.3108 * PHI2 =    -0.3481 RAD AND STRESS * S1 =   540.7238 * S2 =  -605.6371 * S3 =  -163.1943 MPA

*        F1          F2          P1          P2          THT1        THT2        THT3        THT4        P/G1        P/G2        RATIO          T          ITS
  -0.0500     0.1000     2.8131      2.9932     -0.1113     -0.1970     -0.1064     -0.1079      0.9920      1.2953      0.9632    19.3420            6
CORRES WHEEL RATES ARE * C11    11.7049 * C22     9.4829 * C12     5.4853 * C21     5.8600 KN/M
CORRES FREQS ARE * PITCH     0.6975 * BOUNCE     1.2540 HZ        PFB =    1.2531 PFP =     0.6992 HZ        C COEFFS =    -1.5332    -2.6178
CORRES ANGLES * PHI1 =      0.3458 * PHI2 =    -0.4130 RAD AND STRESS * S1 =   601.7220 * S2 =  -718.5466 * S3 =  -238.3021 MPA

*        F1          F2          P1          P2          THT1        THT2        THT3        THT4        P/G1        P/G2        RATIO          T          ITS
   0.0000    -0.1000     2.2967      1.4721      0.0000      0.1978      0.0667      0.0755      0.8099      0.6371      1.3392    14.2229            6
CORRES WHEEL RATES ARE * C11    12.8460 * C22     9.0756 * C12     5.4727 * C21     5.2350 KN/M
CORRES FREQS ARE * PITCH     0.7360 * BOUNCE     1.2551 HZ        PFB =    1.2550 PFP =     0.7362 HZ        C COEFFS =     0.4345     1.2099
CORRES ANGLES * PHI1 =      0.2941 * PHI2 =    -0.2015 RAD AND STRESS * S1 =   494.3321 * S2 =  -350.5829 * S3 =   166.6661 MPA

*        F1          F2          P1          P2          THT1        THT2        THT3        THT4        P/G1        P/G2        RATIO          T          ITS
   0.0000    -0.0500     2.5671      1.8843      0.0000      0.0983      0.0432      0.0368      0.9053      0.8155      1.2517    16.1854            4
CORRES WHEEL RATES ARE * C11    12.5953 * C22     8.8804 * C12     5.5955 * C21     5.2802 KN/M
CORRES FREQS ARE * PITCH     0.7219 * BOUNCE     1.2459 HZ        PFB =    1.2457 PFP =     0.7221 HZ        C COEFFS =     0.6802     0.9801
CORRES ANGLES * PHI1 =      0.3176 * PHI2 =    -0.2623 RAD AND STRESS * S1 =   552.5301 * S2 =  -456.3748 * S3 =    81.2259 MPA
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*      F1          F2          P1          P2          THT1        THT2        THT3        THT4        P/G1        P/G2        RATIO          T          ITS
    0.0000      0.0000      2.8357      2.3108      0.0000      0.0000      0.0000      0.0000      1.0000      1.0000      1.0832     18.2297           0
CORRES WHEEL RATES ARE * C11    12.0358 * C22     8.9934 * C12     5.6679 * C21     5.2324 KN/M
CORRES FREQS ARE * PITCH     0.7035 * BOUNCE     1.2413 HZ        PFB =     1.2414 PFP =     0.7032 HZ        C COEFFS =    -1.3806     0.3934
CORRES ANGLES * PHI1 =       0.3508 * PHI2 =    -0.3239 RAD AND STRESS * S1 =    610.3524 * S2 =   -563.4618 * S3 =      0.0000 MPA

*      F1          F2          P1          P2          THT1        THT2        THT3        THT4        P/G1        P/G2        RATIO          T          ITS
    0.0000      0.0500      3.1046      2.7673      0.0000     -0.0981     -0.0333     -0.0353      1.0948      1.1976      1.1019     20.3828           4
CORRES WHEEL RATES ARE * C11    12.1034 * C22     9.0174 * C12     5.3889 * C21     5.5029 KN/M
CORRES FREQS ARE * PITCH     0.7065 * BOUNCE     1.2431 HZ        PFB =     1.2430 PFP =     0.7065 HZ        C COEFFS =    -0.1385    -0.5219
CORRES ANGLES * PHI1 =       0.3641 * PHI2 =    -0.3867 RAD AND STRESS * S1 =    668.2158 * S2 =   -672.8367 * S3 =    -77.9028 MPA

*      F1          F2          P1          P2          THT1        THT2        THT3        THT4        P/G1        P/G2        RATIO          T          ITS
    0.0000      0.1000      3.3755      3.2725      0.0000     -0.1970     -0.0668     -0.0694      1.1903      1.4162      1.0355     22.6760           6
CORRES WHEEL RATES ARE * C11    11.8556 * C22     9.3563 * C12     5.4579 * C21     5.6902 KN/M
CORRES FREQS ARE * PITCH     0.7041 * BOUNCE     1.2503 HZ        PFB =     1.2498 PFP =     0.7050 HZ        C COEFFS =    -1.2093    -1.8395
CORRES ANGLES * PHI1 =       0.4176 * PHI2 =    -0.4515 RAD AND STRESS * S1 =    726.5310 * S2 =   -785.5893 * S3 =   -153.2123 MPA

*      F1          F2          P1          P2          THT1        THT2        THT3        THT4        P/G1        P/G2        RATIO          T          ITS
    0.0500     -0.1000      2.9719      1.7378      0.1113      0.1978      0.0968      0.1118      1.0480      0.7520      1.4270     18.0211          10
CORRES WHEEL RATES ARE * C11    13.2387 * C22     8.9757 * C12     5.4893 * C21     5.1377 KN/M
CORRES FREQS ARE * PITCH     0.7444 * BOUNCE     1.2597 HZ        PFB =     1.2590 PFP =     0.7455 HZ        C COEFFS =     1.1932     2.2506
CORRES ANGLES * PHI1 =       0.3654 * PHI2 =    -0.2379 RAD AND STRESS * S1 =    635.7050 * S2 =   -413.8493 * S3 =    246.9655 MPA

*      F1          F2          P1          P2          THT1        THT2        THT3        THT4        P/G1        P/G2        RATIO          T          ITS
    0.0500     -0.0500      3.2307      2.1476      0.1113      0.0983      0.0650      0.0734      1.1393      0.9294      1.3344     19.9011           8
CORRES WHEEL RATES ARE * C11    12.9933 * C22     8.7802 * C12     5.4156 * C21     5.1867 KN/M
CORRES FREQS ARE * PITCH     0.7303 * BOUNCE     1.2508 HZ        PFB =     1.2501 PFP =     0.7316 HZ        C COEFFS =     1.4486     2.0280
CORRES ANGLES * PHI1 =       0.3972 * PHI2 =    -0.2989 RAD AND STRESS * S1 =    691.0655 * S2 =   -520.1309 * S3 =    162.1469 MPA

*      F1          F2          P1          P2          THT1        THT2        THT3        THT4        P/G1        P/G2        RATIO          T          ITS
    0.0500      0.0000      3.4684      2.5736      0.1113      0.0000      0.0333      0.0368      1.2302      1.1137      1.2518     21.8577           8
CORRES WHEEL RATES ARE * C11    12.7530 * C22     8.7607 * C12     5.3925 * C21     5.2771 KN/M
CORRES FREQS ARE * PITCH     0.7209 * BOUNCE     1.2468 HZ        PFB =     1.2463 PFP =     0.7217 HZ        C COEFFS =     1.2626     1.4681
CORRES ANGLES * PHI1 =       0.4289 * PHI2 =    -0.3607 RAD AND STRESS * S1 =    746.1840 * S2 =   -627.5337 * S3 =     81.3216 MPA

*      F1          F2          P1          P2          THT1        THT2        THT3        THT4        P/G1        P/G2        RATIO          T          ITS
    0.0500      0.0500      3.7468      3.0315      0.1113     -0.0981      0.0015      0.0016      1.3213      1.3119      1.1768     23.9155           6
CORRES WHEEL RATES ARE * C11    12.5145 * C22     8.9101 * C12     5.4173 * C21     5.4121 KN/M
CORRES FREQS ARE * PITCH     0.7157 * BOUNCE     1.2478 HZ        PFB =     1.2477 PFP =     0.7158 HZ        C COEFFS =     0.6572     0.5690
CORRES ANGLES * PHI1 =       0.4006 * PHI2 =    -0.4236 RAD AND STRESS * S1 =    801.4523 * S2 =   -737.0564 * S3 =      3.6066 MPA

*      F1          F2          P1          P2          THT1        THT2        THT3        THT4        P/G1        P/G2        RATIO          T          ITS
    0.0500      0.1000      4.0077      3.5399      0.1113     -0.1970     -0.0306     -0.0325      1.4133      1.5319      1.1074     26.1033           6
CORRES WHEEL RATES ARE * C11    12.2747 * C22     9.2416 * C12     5.4917 * C21     5.5986 KN/M
CORRES FREQS ARE * PITCH     0.7140 * BOUNCE     1.2546 HZ        PFB =     1.2545 PFP =     0.7141 HZ        C COEFFS =    -0.3949    -0.7110
CORRES ANGLES * PHI1 =       0.4927 * PHI2 =    -0.4884 RAD AND STRESS * S1 =    857.2642 * S2 =   -849.7857 * S3 =    -71.7326 MPA

*      F1          F2          P1          P2          THT1        THT2        THT3        THT4        P/G1        P/G2        RATIO          T          ITS
    0.1000     -0.1000      3.7340      1.9934      0.2241      0.1978      0.1245      0.1468      1.3168      0.8626      1.5176     21.9352          14
CORRES WHEEL RATES ARE * C11    13.9816 * C22     8.8816 * C12     5.5749 * C21     5.1067 KN/M
CORRES FREQS ARE * PITCH     0.7559 * BOUNCE     1.2748 HZ        PFB =     1.2724 PFP =     0.7599 HZ        C COEFFS =     2.5714     3.8198
CORRES ANGLES * PHI1 =       0.4504 * PHI2 =    -0.2728 RAD AND STRESS * S1 =    783.6026 * S2 =   -474.7049 * S3 =    324.2049 MPA

*      F1          F2          P1          P2          THT1        THT2        THT3        THT4        P/G1        P/G2        RATIO          T          ITS
    0.1000     -0.0500      3.9848      2.4014      0.2241      0.0983      0.0943      0.1088      1.4052      1.0392      1.4193     23.7337          12
CORRES WHEEL RATES ARE * C11    13.7343 * C22     8.6862 * C12     5.5033 * C21     5.1604 KN/M
CORRES FREQS ARE * PITCH     0.7417 * BOUNCE     1.2661 HZ        PFB =     1.2637 PFP =     0.7459 HZ        C COEFFS =     2.8270     3.5879
CORRES ANGLES * PHI1 =       0.4806 * PHI2 =    -0.3343 RAD AND STRESS * S1 =    836.2288 * S2 =   -581.5987 * S3 =    240.1634 MPA

*      F1          F2          P1          P2          THT1        THT2        THT3        THT4        P/G1        P/G2        RATIO          T          ITS
    0.1000      0.0000      4.2351      2.8274      0.2241      0.0000      0.0641      0.0724      1.4935      1.2236      1.3320     25.6053          10
CORRES WHEEL RATES ARE * C11    13.4927 * C22     8.6647 * C12     5.4833 * C21     5.2546 KN/M
CORRES FREQS ARE * PITCH     0.7326 * BOUNCE     1.2620 HZ        PFB =     1.2601 PFP =     0.7359 HZ        C COEFFS =     2.6431     3.0273
CORRES ANGLES * PHI1 =       0.5108 * PHI2 =    -0.3963 RAD AND STRESS * S1 =    888.7469 * S2 =   -689.4361 * S3 =    159.8924 MPA

*      F1          F2          P1          P2          THT1        THT2        THT3        THT4        P/G1        P/G2        RATIO          T          ITS
    0.1000      0.0500      4.4866      3.2872      0.2241     -0.0981      0.0338      0.0374      1.5821      1.4225      1.2530     27.5723          10
CORRES WHEEL RATES ARE * C11    13.2537 * C22     8.8101 * C12     5.5123 * C21     5.3926 KN/M
CORRES FREQS ARE * PITCH     0.7279 * BOUNCE     1.2625 HZ        PFB =     1.2615 PFP =     0.7297 HZ        C COEFFS =     2.0420     2.1362
CORRES ANGLES * PHI1 =       0.5411 * PHI2 =    -0.4594 RAD AND STRESS * S1 =    941.5234 * S2 =   -799.2329 * S3 =     82.5225 MPA
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*      F1          F2          P1          P2          THT1        THT2        THT3        THT4        P/G1        P/G2        RATIO          T          ITS
    0.0500      0.0500      3.7468      3.0315      0.1113     -0.0981      0.0015      0.0016      1.3213      1.3119      1.1768    23.9155            6
CORRES WHEEL RATES ARE * C11    12.5145 * C22     8.9101 * C12     5.173 * C21     5.4121 KN/M
CORRES FREQS ARE * PITCH      0.7157 * BOUNCE     1.2478 HZ      PFB =     1.2477 PFP =     0.7158 HZ      C COEFFS =      0.6572     0.5690
CORRES ANGLES * PHI1 =      0.4006 * PHI2 =     -0.4236 RAD AND STRESS * S1 =    801.4523 * S2 =   -737.0564 * S3 =      3.6066 MPA

*      F1          F2          P1          P2          THT1        THT2        THT3        THT4        P/G1        P/G2        RATIO          T          ITS
    0.0500      0.1000      4.0077      3.5399      0.1113     -0.1970     -0.0306     -0.0325      1.4133      1.5319      1.1074    26.1033            6
CORRES WHEEL RATES ARE * C11    12.2747 * C22     9.2416 * C12     5.4917 * C21     5.5986 KN/M
CORRES FREQS ARE * PITCH      0.7140 * BOUNCE     1.2546 HZ      PFB =     1.2545 PFP =     0.7141 HZ      C COEFFS =     -0.3949    -0.7110
CORRES ANGLES * PHI1 =      0.4927 * PHI2 =     -0.4884 RAD AND STRESS * S1 =    857.2642 * S2 =   -849.7857 * S3 =    -71.7326 MPA

*      F1          F2          P1          P2          THT1        THT2        THT3        THT4        P/G1        P/G2        RATIO          T          ITS
    0.1000     -0.1000      3.7340      1.9934      0.2241      0.1970      0.1245      0.1468      1.3168      0.8626      1.5176    21.9352           14
CORRES WHEEL RATES ARE * C11    13.9816 * C22     8.8816 * C12     5.5749 * C21     5.1067 KN/M
CORRES FREQS ARE * PITCH      0.7559 * BOUNCE     1.2748 HZ      PFB =     1.2724 PFP =     0.7599 HZ      C COEFFS =      2.5714     3.8198
CORRES ANGLES * PHI1 =      0.4804 * PHI2 =     -0.2728 RAD AND STRESS * S1 =    783.6026 * S2 =   -474.7049 * S3 =    324.2049 MPA

*      F1          F2          P1          P2          THT1        THT2        THT3        THT4        P/G1        P/G2        RATIO          T          ITS
    0.1000     -0.0500      3.9848      2.4014      0.2241      0.0984      0.0943      0.1088      1.4052      1.0392      1.4193    23.7337           12
CORRES WHEEL RATES ARE * C11    13.7343 * C22     8.6862 * C12     5.5033 * C21     5.1604 KN/M
CORRES FREQS ARE * PITCH      0.7417 * BOUNCE     1.2661 HZ      PFB =     1.2637 PFP =     0.7459 HZ      C COEFFS =      2.8270     3.5879
CORRES ANGLES * PHI1 =      0.4806 * PHI2 =     -0.3343 RAD AND STRESS * S1 =    836.2288 * S2 =   -581.5987 * S3 =    240.1634 MPA

*      F1          F2          P1          P2          THT1        THT2        THT3        THT4        P/G1        P/G2        RATIO          T          ITS
    0.1000      0.0000      4.2351      2.8274      0.2241      0.0000      0.0641      0.0724      1.4935      1.2236      1.3320    25.6053           10
CORRES WHEEL RATES ARE * C11    13.4927 * C22     8.6647 * C12     5.4833 * C21     5.2546 KN/M
CORRES FREQS ARE * PITCH      0.7326 * BOUNCE     1.2620 HZ      PFB =     1.2601 PFP =     0.7359 HZ      C COEFFS =      2.6431     3.0273
CORRES ANGLES * PHI1 =      0.5108 * PHI2 =     -0.3963 RAD AND STRESS * S1 =    888.7469 * S2 =   -689.4381 * S3 =    159.8924 MPA

*      F1          F2          P1          P2          THT1        THT2        THT3        THT4        P/G1        P/G2        RATIO          T          ITS
    0.1000      0.0500      4.4866      3.2872      0.2241     -0.0981      0.0338      0.0374      1.5821      1.4225      1.2530    27.5723           10
CORRES WHEEL RATES ARE * C11    13.2537 * C22     8.8101 * C12     5.5123 * C21     5.3926 KN/M
CORRES FREQS ARE * PITCH      0.7279 * BOUNCE     1.2625 HZ      PFB =     1.2615 PFP =     0.7297 HZ      C COEFFS =      2.0420     2.1362
CORRES ANGLES * PHI1 =      0.5411 * PHI2 =     -0.4594 RAD AND STRESS * S1 =    941.5234 * S2 =   -799.2329 * S3 =     82.5225 MPA

*      F1          F2          P1          P2          THT1        THT2        THT3        THT4        P/G1        P/G2        RATIO          T          ITS
    0.1000      0.1000      4.7411      3.7994      0.2241     -0.1970      0.0031      0.0033      1.6719      1.6442      1.1803    29.6613            8
CORRES WHEEL RATES ARE * C11    13.0141 * C22     9.1351 * C12     5.5923 * C21     5.5814 KN/M
CORRES FREQS ARE * PITCH      0.7271 * BOUNCE     1.2684 HZ      PFB =     1.2682 PFP =     0.7275 HZ      C COEFFS =      0.9969     0.8737
CORRES ANGLES * PHI1 =      0.5718 * PHI2 =     -0.5242 RAD AND STRESS * S1 =    994.9341 * S2 =   -912.0706 * S3 =      7.3209 MPA
```

C_{12} and C_{21} are linear, which is not the case. The difference is less than 2%.

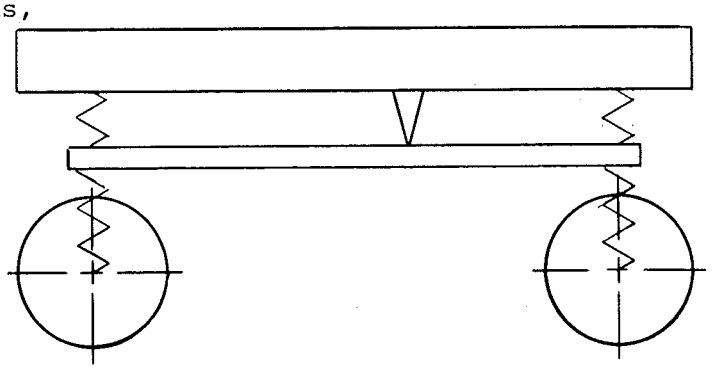
The front and rear wheel displacements relative to the vehicle body F_1 and F_2 are now iterated, from -100 to 100 mm in steps of 50 mm for this printout. For each of these 25 permutations all the suspension, vehicle and mechanical details discussed above are calculated and printed out.

A complete printout of the program with flowcharts and a brief description thereof is included in Appendix 2.4.2.

2.4.2 Discussion of Results

Table 2.4.1 lists selected results extracted from the computer printout. The springrates C_{11}, C_{22}, C_{12} and C_{21} are those referred to in equation (2.2.1).

From table 2.4.1 the rising springrate effect (section 2.4.1) is apparent. For a conventional car, for which $C_{12} = C_{21} = 0$, with rising rate suspension, using for example a coil spring made from tapered coil wire, the maximum values of C_{11} would occur at 100, 0 and 100,000 and for C_{22} at 0, 100 and 100,100. For the interconnected suspension systems shown in figures 2.2.1 and 2.3.1 as well as for the Packard system, maximum C_{11} and C_{22} would occur at the double bump condition of 100, 100. For this interconnected suspension, maximum C_{11} occurs at 100, -100 and maximum C_{22} at -100, 100, a feature it shares in common with the Citroen 2CV. With reference to figure 2.2.1, this suspension system would be represented as,



For a car with conventional suspension the ratio of dynamic to static wheel loadings is a maximum at both single and double bump conditions. For example, the Porsche 928 has maximum values of 1.48 and 1.62 for the front and rear respectively. For this car, with interconnected suspension, the maximum wheel loading ratios of 1.67 and 1.64 front and rear respectively, occur at the double bump condition of 100,100,

TABLE 2.4.1 Selected Printout Results

Wheel Displacement Front, Rear	0, 0	100, 100	100, -100	-100, 100	0, 100	100, 0
C_{11} kN/m	12.04	13.01	13.98	11.81	11.86	13.49
C_{22} kN/m	8.99	9.14	8.88	9.63	9.36	8.66
C_{12} kN/m	5.67	5.59	5.57	5.57	5.46	5.48
C_{21} kN/m	5.23	5.58	5.12	6.12	5.69	5.25
P_1/G_1	1.0	1.67	1.32	0.811	1.19	1.49
P_2/G_2	1.0	1.64	0.863	1.17	1.42	1.22
Bounce Frequency cpm	74.5	76.1	76.5	75.9	75.0	75.7
Pitch Frequency cpm	42.2	43.6	45.4	41.7	42.2	44.0
Cable T kN	18.2	29.7	21.9	16.1	22.7	25.6
C_1 Stress S_1 MPa	610	995	784	482	727	889
C_2 Stress S_2 MPa	-563	-912	-475	-648	-786	-689
C_3 Stress S_3 MPa	0	7	324	-328	-153	160

in common with the conventional suspension. However, in contrast to the conventional suspension, these reduce for the single bump conditions of 100, 0 and 0, 100 and then reduce even further for the one wheel bump and other wheel rebound conditions of 100, -100 and -100, 100, illustrating clearly the mechanisism for reduced pitching.

It should be observed, that, although the springrates of C_{11} and C_{22} are a maximum at 100, -100 and -100, 100 respectively, the effect of interconnection reduces the actual wheel loading of the wheel in bounce and increases the actual wheel loading of the wheel in rebound compared to a car with conventional suspension and thus reduces the pitching .

The object of this interconnected suspension is to give large car ride quality to what is basically a short car. With a wheelbase of 2230 mm the Fiat 127 at 2223 is shorter, while others like the 2CV, VW Golf and Beetle at 2400, the Escort at 2407, the Allegro at 2438, the Fiat 128 at 2448 and the Avenger at 2489 are all longer.

The frequencies of cars like the Rover 3500 and the BMW 728 where used as a guide, while those of the Allegro were considered too high. With anti-dive and self-levelling, frequencies as low as those used by Packard cars in 1956 could be considered (see section 2.1).

Self-levelling and anti-dive (Chapter 3) are incorporated. The self-levelling is achieved by using an air-spring/damper unit between the rear suspension arm and the body. This augments the suspension system, when additional loads are placed in the rear of the vehicle, by increasing C_{22}.

Early analysis had shown, that, given the geometric restrictions of the vehicle, namely the relatively narrow track and the relatively long rear trailing arm required to accommodate the motor and pulleys and the matching long effective front leading arm, that the torsion bars would be highly stressed. The torsion bars would require through hardening, shot-peening and presetting (or scragging), SAE(1947), which would allow 965 - 1035 MPa maximum working stress, depending on the amount of settling of the vehicle that would be acceptable.

Analysis of torsion bars on a number of production cars that use torsion bar suspension was then carried out. It was found that the short-series torsion bars used on the rear of certain VW Beetles would not only fit into the Electric Town Car, but had an acceptable torsion bar spring-rate and worked at similar stresses to that required in the Electric Town Car. It made economic sense to use these bars.

The computer program was altered to analyse a series of different auxiliary torsion bars on one printout, in combination with the VW torsion bars. The resultant series of pitch and bounce frequencies was examined, and with due consideration for the dynamic to static wheel loadings, the actual pitch and bounce frequencies, the brake dive angles, the self-levelling facility, the torsion bar stresses and so on, a pitch and bounce frequency combination of 42 and 75 cycles per minute respectively was chosen.

A German reference, Gross (1966) was used for the torsion moduli of the VW bars. A maximum stress of 980 MPa was also recommended for the pure torsion condition.

Torsion bars are normally loaded in one direction. The auxiliary torsion bar is loaded in both directions, the maximum recommended stress for which is 340 to 390 MPa, Gross (1966). From table 2.4.1 it can be seen that the maximum working stress is 328 MPa. It was not therefore considered prudent to use this auxiliary torsion bar as a method of self-levelling.

The steel used for the auxiliary torsion bar was XK92631 which is nominally equivalent to En45A. After machining, the bars were oil quenched at between 920 and 930^{o}C and tempered at 500^{o}C as recommended in "The Mechanical and Physical Properties of the British En Steels".

The cable tension has a maximum value of 29.7 kN at the double bump configuration of 100, 100. This tension was then used to select the cable size and to design the Glacier DU suspension pivot bearings. The chassis load for the two cables is 59.4 kN.

CHAPTER 3

SUSPENSION GEOMETRY

3.1 PITCH CHARACTERISTICS

While interconnected suspension can improve ride quality by reducing the pitch frequency and hence pitch acceleration, it can, depending on the choice of basic and interconnecting stiffnesses, produce the need for self-levelling and anti-dive geometry.

The choice of frequencies and related springrates for this vehicle are such that both self-levelling and anti-dive geometry are required. From section 2.2.2, the body tilt for the placing of 750 N over the rear axle is approximately 2.2^{o} without self-levelling and for a 0.6 g braking deceleration -7.3^{o} without anti-dive geometry.

Self-levelling is achieved by using air-springs at the rear as discussed in section 2.4.2. The control for levelling is manual, using an indicator attached to one pitch control torsion bar and a small electric air compressor.

With reference to figure 3.1.1, pT is the braking force on the front wheels and (1-p)T on the rear wheels, where T is the total braking force. With outboard brakes front and rear and with the suspension geometry such that these forces are purely horizontal, then the braking moment is (pT + (1-p)T)h = Th, where h is the height of the centre of gravity above ground level. This is the condition of zero anti-dive and the body tilt is given by equation 2.2.16.

If the suspension geometry were such that the front and rear braking forces acted through the transverse axis of the centre of gravity, figure 3.1.2, then the moment would be zero and the body would not tilt. This is the condition of 100% anti-dive. If the braking forces front and rear intersected the vertical line through the centre of gravity at 0.3 h, figure 3.1.3, the braking moment would be 0.7 Th. This is the condition of 30% anti-dive. The percentage of anti-dive front and rear are not necessarily the same.

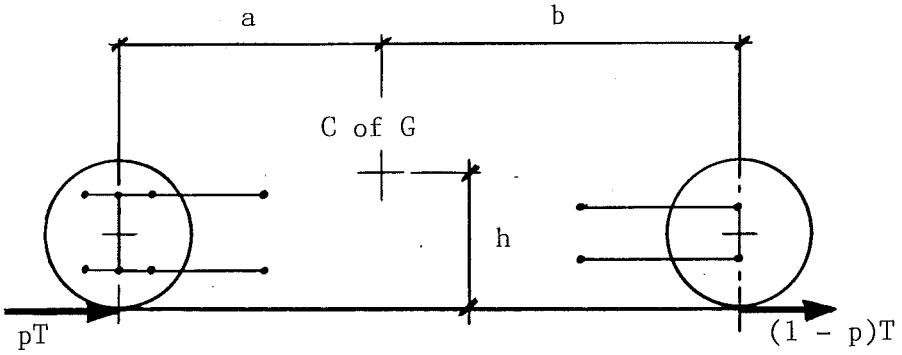


Figure 3.1.1. No Anti-Dive.

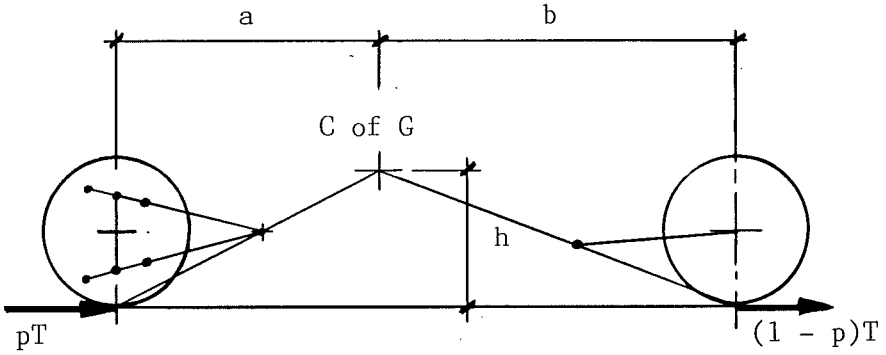


Figure 3.1.2. 100 % Anti-Dive.

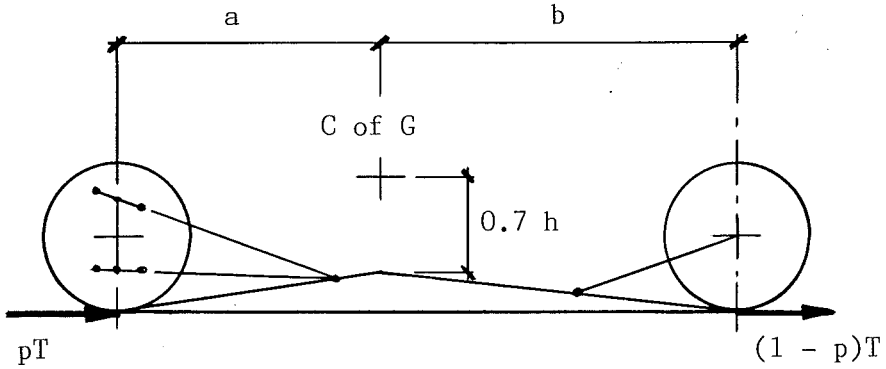


Figure 3.1.3. 30 % Anti-Dive.

Consider the front suspension geometry, figure 3.1.4, where a is the distance of the centre of gravity behind the front axle centreline, and the angle of the line of the front braking force is θ_f. The moment due to the front wheels braking force is

$$M_f = pT(h - a \tan \theta_f) \tag{3.1.1}$$

which can be written

$$M_f = pTh\left(1 - \frac{a \tan \theta_f}{h}\right)$$

so that if $\frac{a \tan \theta_f}{h} = 0.3$, then $M_f = 0.7\ pTh$ and there exists 30% anti-dive at the front.

Similarly for the rear, where b is the distance from the centre of gravity to the rear axle centre line

$$M_r = (1-p)\ T\ (h - b \tan \theta_r) \tag{3.1.2}$$

Combining 3.1.1 and 3.1.2 to give the total moment

$$M = T(h - pa \tan \theta_f - (1-p)\ b \tan \theta_r) \tag{3.1.3}$$

Substituting equation (3.1.3) for Th in equation (2.2.16), the body tilt angle becomes

$$\Delta\psi = - \frac{C_{11} + C_{22} + C_{12} + C_{21}}{C_{11}\ C_{22} - C_{12}\ C_{21}} \quad \frac{T(h - pa \tan \theta_f - (1-p)\ b \tan \theta_r)}{L^2} \tag{3.1.4}$$

If $\theta_f = \theta_r = 0$, the conditions for zero anti-dive, then equation (3.1.4) reduces to equation (2.2.16).

From figure 3.1.5, the angle of the line of the front braking force θ_f is 8.6°. With a = 1.001 m and h = 0.491 m, figure 3.1.7, this corresponds to 31% anti-dive. Nominally 30% front anti-dive is the largest percentage used. Above this, variations in castor angle with wheel displacement can become a problem.

From figure 3.1.6, the angle of the line of the rear braking force $\theta_r = 27.9^\circ$. With b = 1.229 m and h = 0.491 m, figure 3.1.7, this corresponds to 133% anti-dive. This rear suspension is such that a rear wheel braking force will produce a negative rear pitch moment and hence squatting, a feature shared with the Leyland Hydrolastic and Hydragas cars and most easily demonstrated by applying the handbrake while travelling slowly in

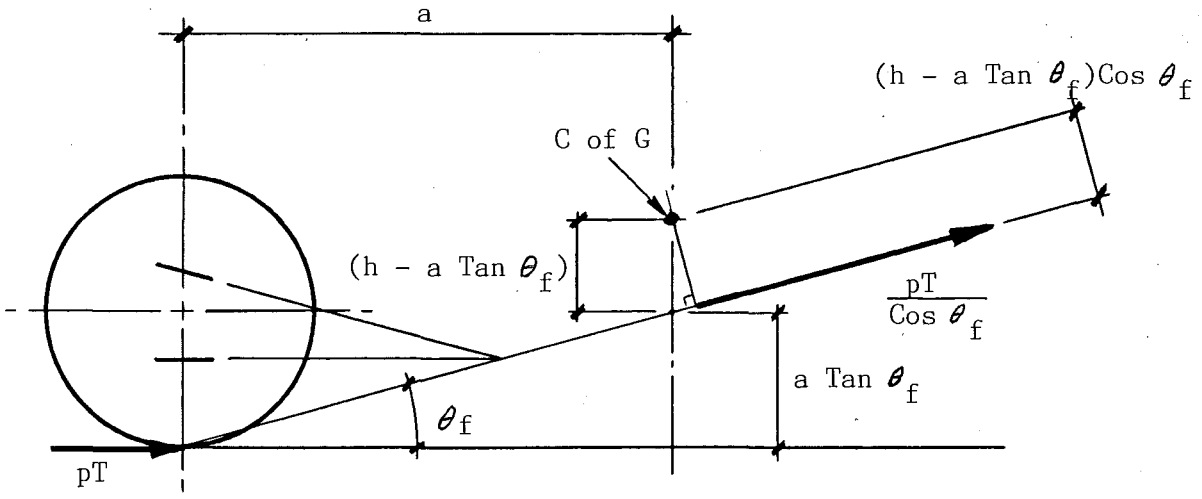


Figure 3.1.4. Side Elevation of Front Suspension Geometry.

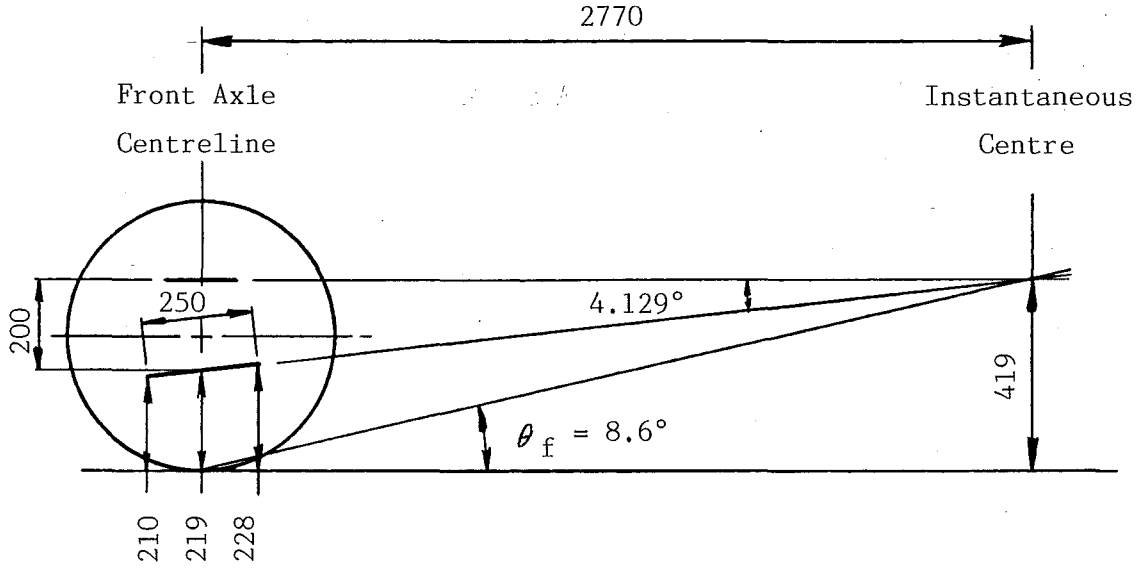


Figure 3.1.5. As Built Front Wishbone Side Elevation Geometry.
Based on Drawing 8802 of Appendix 6.1.

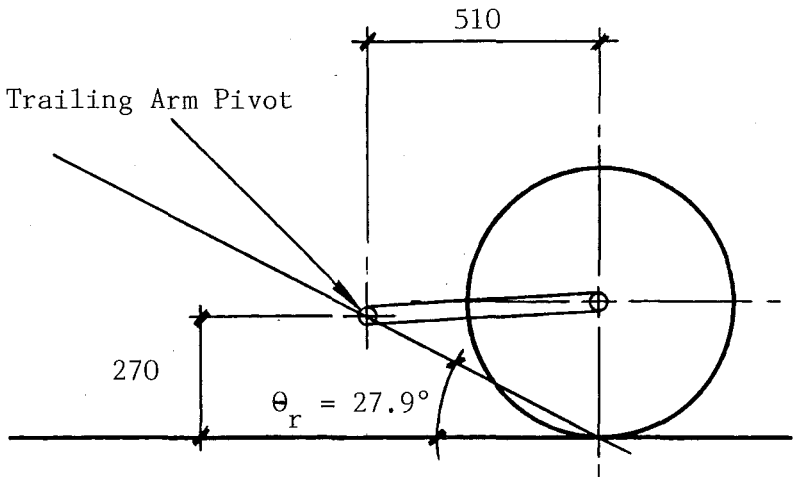


Figure 3.1.6. Rear Suspension Trailing Arm, Side Elevation.

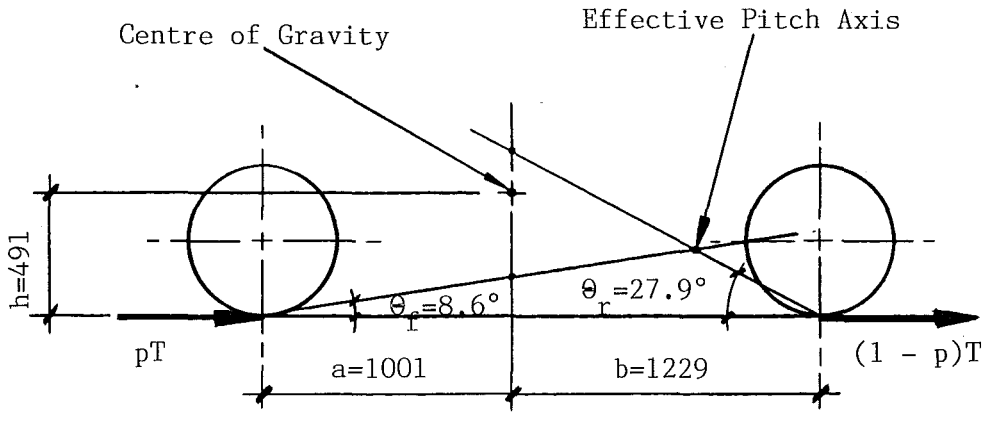


Figure 3.1.7. Vehicle Side Elevation Showing the Line of Action of Braking Forces.

either direction. Due to the short trailing arm length, the percentage anti-dive varies with wheel displacement, for example at 100,100 front, rear wheel displacement conditions, θ_r is 18.43^o which corresponds to 105% anti-dive.

Substituting values for C_{11}, C_{22}, C_{12} and C_{21} for the 0,0 position from table 2.4.1, but doubling them to convert from single wheelrates to vehicle rates, and a, b and h from figure 3.1.7, equation (3.1.4) becomes,

$$\Delta\psi = -2.341(10)^{-3}\ T(0.4993p - 0.1597) \text{ degrees} \qquad (3.1.5)$$

The Electric Town Car has regenerative braking on the rear drive wheels only and no other rear wheel braking except the handbrake. The electrical system is set up such that initial depression of the brake pedal produces regenerative rear wheel braking only, hence p = 0 for small braking values. Depressing the pedal further, brings in the conventional front disc brakes in addition to the rear wheel regenerative braking. The proportion of braking effort from the front wheels, p, will increase from zero to perhaps 0.8. From equation (3.1.5), at p = 0.32 $\Delta\psi$ = 0, so that below p = 0.32 the car will exhibit a tail down attitude and above p = 0.32 a nose down attitude. If the electric circuit breaker were to trip under heavy braking, there would be no rear wheel braking and p = 1. Table 3.1.1 lists body tilt angles for different values of T and p.

TABLE 3.1.1 Body Tilt Angle (degrees)

p \ T/Mg	0.1	0.2	0.3	0.4	0.5	0.6	0.7
0	0.39	0.77	1.16	1.54			
0.5	-0.22	-0.43	-0.65	-0.87	-1.08	-1.3	-1.52
0.8	-0.58	-1.16	-1.73	-2.31	-2.89	-3.47	-4.05
1.0	-0.82	-1.64	-2.46	-3.28	-4.09	-4.91	-5.73

The angle of -3.47 degrees at T = 0.6 Mg and p = 0.8 is on the upper limit of being acceptable for body tilt of a conventional car. The emergency condition of p = 1 produces excessive body tilt angle, reduced by applying the handbrake during the said emergency. Should the body tilt angles be uncomfortably excessive in practice, the rear drum brakes can be connected into the hydraulic brake in the conventional manner.

With the electric motor on the rear suspension trailing arm (Chapter 4), tractive forces are transmitted through the trailing arm pivots, and the same theory applies for anti-squat as it did for anti-dive. Equation (3.1.4) applies, with p=0 for rear wheel drive and the tractive force F = -T, so that

$$\Delta\psi = \frac{C_{11} + C_{22} + C_{12} + C_{21}}{C_{11} C_{22} - C_{12} C_{21}} \quad \frac{F(h - b \tan \theta_r)}{L^2} \qquad (3.1.6)$$

Acceleration tests with the Mk 1 Electric Car in its final state of tune in May 1978, produced an acceleration force of 1287 N. The same drive system is to be used for this vehicle. An acceleration force of 1340 N would produce an acceleration of 0.13 g and the body tilt would be -0.5 degrees.

3.2 ROLL CHARACTERISTICS

Just as the vehicle body exhibited pitching about an effective pitch axis (figure 3.1.7) or transverse roll axis (Steeds, 1958) due to braking forces, so too will it exhibit roll about a longitudinal roll axis due to cornering forces. While the effective pitch axis is horizontal, that is, the same height on both sides of the vehicle, the longitudinal roll axis generally is inclined to the ground, and is found by joining the individual front and rear suspension roll centres. The position of these roll centres is established from the linkage geometry in the same way as the effective pitch axis in figure 3.1.7, but by considering the suspension geometry in end elevation. The geometry left and right is generally symmetrical, so that, unlike the pitch axis, the roll centre is on the vehicle centreline.

The rear suspension comprises a pair of independent trailing arms, the pivot axis for which is horizontal with the vehicle body in end elevation and perpendicular with the body centreline in planview. The rear wheels move vertically on a single wheel bump, the roll centre is therefore at ground level. The camber angle of both rear wheels is equal in magnitude to the body roll angle at all times.

The front suspension comprises a conventional wishbone layout, chosen because: 1) anti-dive geometry can be incorporated; 2) the roll-centre height can be positioned where desirable; 3) the layout gives clearance to store the front batteries and to incorporate the interconnected

suspension leading arm; 4) dampers and anti-roll bars can be fitted easily and 5) the steering connections can be made to minimise toe-in and toe-out effects with wheel movement.

The total lateral weight transfer from the inside tyres to the outside tyres, regardless of suspension design, is Fh/t, where F is the cornering force, h the height of the centre of gravity above groundlevel and t the average track width. This total weight transfer comprises weight transfer due to the unsprung weight, front and rear, weight transfer due to the sprung weight acting at the roll centres and weight transfer due to body roll of the sprung weight.

With a nominal front weight bias, the handling characteristics of slight understeer will be produced if the weight transfer is made proportional to the front and rear weight distribution. The front and rear wheelrates are proportioned to match the weight distribution with the interconnected suspension design. From section 2.2.1, the front and rear wheel reactions are:

$$\left.\begin{aligned} P_1 &= C_{11}f_1 + C_{12}f_2 \\ P_2 &= C_{21}f_1 + C_{22}f_2 \end{aligned}\right\} \qquad (2.2.1)$$

For roll conditions, with no pitching, $f_1 = f_2$ and equations (2.2.1) become

$$P_1 = (C_{11} + C_{12})f_1$$

$$P_2 = (C_{21} + C_{22})f_2$$

so that $(C_{11} + C_{12})$ and $(C_{21} + C_{22})$ become the effective wheelrates, front and rear respectively.

The roll stiffness for the front is

$$\frac{\pi}{180} \cdot \frac{t_f^{\,2}(C_{11} + C_{22})}{2} = 254 \text{ Nm/degree}$$

and for the rear

$$\frac{\pi}{180} \quad \frac{t_r^{\,2}(C_{21} + C_{22})}{2} = 191 \text{ Nm/degree}$$

where $t_f = 1.282$ m and $t_r = 1.240$ m are the respective vehicle tracks. Wheelrates C_{11}, C_{12}, C_{21} and C_{22} are obtained from table 2.4.1 using the 0,0 bounce condition. The front roll stiffness is further increased by

the installation of an anti-roll bar. The front suspension has the higher roll stiffness, so that it will receive the larger portion of weight transfer due to body roll.

The weight transfer due to the unsprung weight is greater at the rear. The motor and steel trailing arm have an unsprung weight of approximately 50 kg per side, compared to 15 kg per side at the front.

Weight transfer due to the sprung weight acting through the rear roll centre is zero as the roll centre height is at ground level. The roll centre at the front needs to be approximately at ground level also, to retain the balance. If the front roll centre height is increased a little above ground level, this increases the weight transfer due to the sprung weight acting at the roll centre, but decreases the weight transfer due to body roll, which would be beneficial as the front roll stiffness is greater than the rear. A front roll centre height of 35 mm above ground level was chosen as being consistent with the above and producing acceptable camber variation with bump and roll, without deviating too much from the characteristics of the rear suspension. The chosen layout fits well into the confined space of the front of the body used for the Electric Town Car.

The roll axis is therefore inclined to the ground, being 35 mm above ground at the front and at ground level at the rear. Normally the axis would be higher at the rear. The height of the roll axis at the centre of gravity position is 19.5 mm. The weight transfer due to body roll for a 0.5 Mg cornering force, with h = 0.491 m and M = 1050 kg, is 2428 Nm, and with a combined front plus rear roll stiffness, ignoring the front anti-roll bar, of 445 Nm/degree, the body roll is an acceptable 5.46 degrees.

In order to ascertain the roll centre height and camber variation of the front wishbone suspension for all bounce and rebound conditions a computer program WISHANAL, standing for "Wishbone Analysis", was written. The coordinates of the suspension are inputted as a set of seven coordinates, figure 3.2.1, together with the bounce and rebound limits required to be analysed. The program calculates and prints all the relevant lengths and angles, specifically isolating the length of the upper and lower suspension arms as well as the kingpin angle. A sample output printout is included here, while a complete printout of the program with flowcharts and a brief description thereof is included in Appendix 3.2.

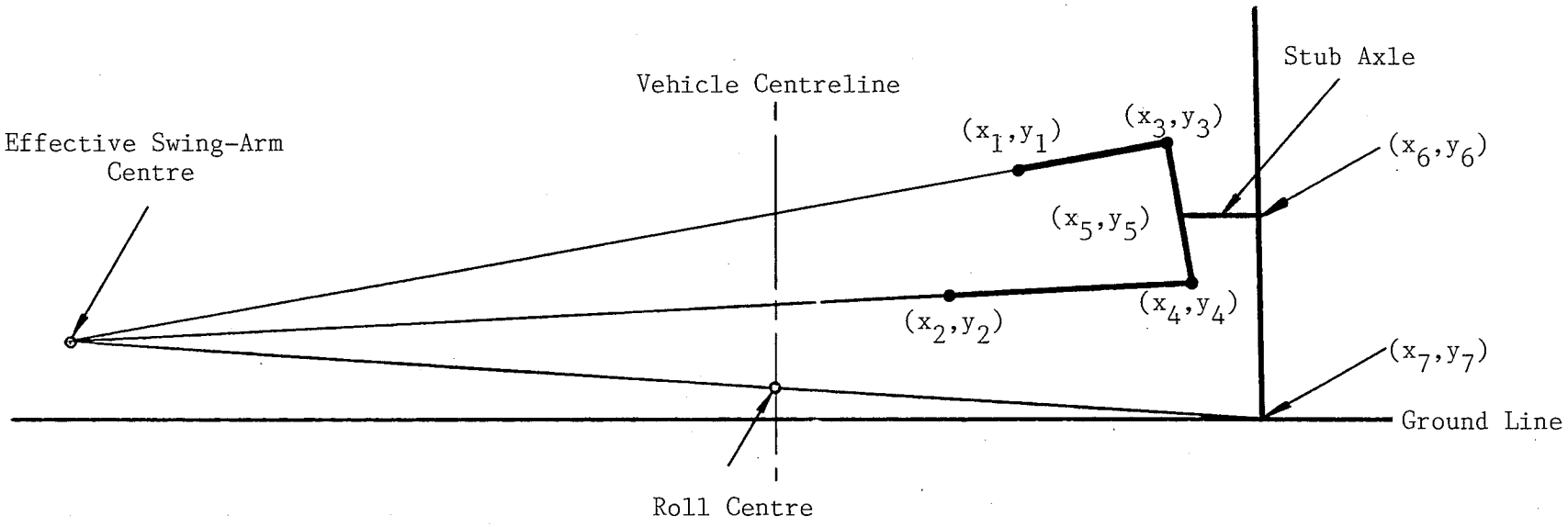


Figure 3.2.1. Wishbone Suspension, Front Elevation.

PRELIMINARY ANALYSIS OF WISHBONE SUSPENSION GEOMETRY

	X-COORD	Y-COORD
1	330.0	419.0
2	262.0	219.0
3	522.0	414.0
4	555.0	207.0
5	0.0	0.0
6	632.0	280.0
7	637.0	0.0

TO CALCULATE FOR BOUNCE 100 AND REBOUND 100

LENGTH OF UPPER ARM IS 192.065

LENGTH OF LOWER ARM IS 293.246

LENGTHS	L12	L13	L24	L34	L35	L36	L37
	211.244	192.065	293.246	209.614	666.243	173.367	429.675

ANGLES	AX12	AX13	AX24	AX34	AX35	AX36	AX37
	-1.890	-0.026	-0.041	-1.413	-2.471	-0.883	-1.300

ANGLES	A435	A436	A437
	-1.058	0.529	0.113

KINGPIN ANGLE = 8.035

SCRUB IS POSITIVE OUTWARDS

	K	Y7	RY	RY-Y7	CAMBER	SCRUB	SARM	RATIO
*	0	0.000	35.418	35.418	-1.023	0.000	13887.968	10.901
**	25	25.521	34.152	8.631	-1.280	0.880	3598.710	2.825
*	50	51.705	34.223	-17.483	-1.851	0.691	2067.577	1.623
*	75	78.701	37.287	-41.413	-2.771	-0.571	1413.204	1.109
*	100	106.821	45.553	-61.268	-4.021	-2.866	1031.493	0.810
**	0	0.000	35.418	35.418	-1.023	0.000	13887.968	10.901
**	-25	-24.890	36.360	61.250	-1.078	-1.896	6399.862	5.023
**	-50	-49.068	34.557	83.625	-1.481	-4.674	2258.269	1.773
**	-75	-72.290	25.026	97.315	-2.321	-8.043	1159.415	0.910
*-	100	-93.974	-6.404	87.570	-3.786	-11.351	616.642	0.484

For each bump and rebound condition specified, the following is calculated and printed.

K, the deviation in y-coordinate of the upper-outboard balljoint (x_3, y_3) from its static condition. The upper balljoint was chosen rather than the more logical bottom balljoint because the program has the facility for the lower arm, defined by (x_2, y_2) and (x_4, y_4), to be horizontal or any other fixed angle, in all conditions. The lower arm can physically be a leading arm. Bounce is positive, rebound is negative.

Y7, the y-coordinate of the tyre/ground contact point.

RY, the instantaneous roll-centre height relative to the vehicle.

RY-Y7, the instantaneous roll-centre height relative to the tyre/ground contact point.

CAMBER, the camber angle of the tyre, that is the angle the wheel centreline makes to the vertical. Negative camber is the top of the tyre leaning towards the centre of the vehicle. CAMBER assumes no body roll.

SCRUB, the lateral (sideways) displacement of the bottom of the tyre (road interface) from its static position. Scrub is positive outwards.

SARM, the instantaneous swingarm length, that is the length of the line, parallel to the stub axle, from the instantaneous swingarm centre (where the top and bottom suspension arms geometrically intersect) to the wheel centreline.

$$\text{RATIO} = \frac{\text{SARM}}{\text{TRACK}} = \frac{\text{swing-arm length}}{\text{distance between front wheel centres}}$$

The camber variation with bounce is deliberately small for the outside more highly loaded wheel, so as to match closely the characteristics of the rear outside wheel, where there is no camber variation with bounce. For the 0.5 Mg corner, where the body roll is approximately 5.46 degrees, the outer rear wheel has a positive camber of 5.46 degrees and the inner rear wheel has a negative camber of 5.46 degrees, while the outer front wheel, with an effective wheel displacement of 61 mm, will have a positive camber of 3.3 degrees and the inner front wheel a negative camber of 7.4

degrees. The values of scrub, effective swing arm length and its ratio to the vehicle track, complete the picture of how the particular suspension geometry is working.

It is possible to carry out a more complete analysis of weight transfer and body roll, (Bastow, 1980), and steady-state cornering behaviour, (Radt and Pacejka, 1963), (Pacejka, 1973), but these would require more detailed information on individual front and rear centre of gravity positions, roll moments of inertia, tyre characteristics (Gough, 1954), (Setright 1972) and so on. The economics of using the existing anti-roll bar and some as built suspense tuning and development, if necessary, provides the solution.

CHAPTER 4

REAR SUSPENSION TRAILING ARM

4.1 DRIVE SYSTEM

It had been proposed, Vandendungen (1976), to use a single Synchro-belt drive for each of the two driven rear wheels, with the motor and pulleys attached to the rear suspension trailing arm. A double-reduction belt drive had been successfully used on the earlier Electric Car prototype, with occasional belt-tooth jumping during regenerative braking.

The existing system was analysed, working backwards to establish the appropriate belt tensions and service factors actually being used. These tensions and factors were applied to the single belt drive system indicating unacceptable belt widths of between 76 and 90 mm depending on the actual ratio chosen. It was known that stronger belts were available, Hartley 1976. Powergrip HTD Belts from Uniroyal were investigated indicating a width of 50 mm might be possible if non-standard tensions and service factors were used. Technical correspondence was entered into with Uni-royal in England, concerning the proposed 50 mm belt and is included in Appendix 4.1. From the correspondence of Powergrip Industries in Australia, 17 July 78, it can be seen that the cheapest option, with duty excluded, was $970.84 for a set of steel pulleys and a pair of 1280 mm belts. The equivalent using three-strand chain and standard steel and cast iron pulleys was $190, chain was subsequently used.

4.2 SUSPENSION ARM

The rear suspension arm as originally proposed, (Vandendungen, 1976), was to comprise an aluminium mounting plate with a cast or fabricated alum-inium or steel beam structure around the back of the motor, extending from the stub axle to the inner end of the torsion bar mounting tube. With the restriction of the body used for this vehicle, with its narrow rear track, plus the need for the provision of wider belts (although chains were subse-quently chosen), a small increase in tunnel/backbone width and optional use of the more powerful but longer MA112M electric motor (standard motor is MA100L) as well as a preference to be able to remove the motors from the

car without partially dismantling the suspension arm, all meant that there was insufficient space between the back of the motor and the chassis for such a beam. This led to the development of an all-plate type rear suspension trailing-arm, drawings 7937/1 and 7956 of Appendix 6.1, made from steel for ease of manufacture and for stiffness, and of minimum depth to allow the arm, complete with motor, to fit under the rear seats in the maximum bump configuration.

With a rear track for this vehicle, of 1240 mm, all the cars used for comparison in section 2.4.2 have a wider rear track. The Citroen 2CV at 1260 mm is only 20 mm wider, while the Escort, Fiat 127, Avenger, Fiat 128, VW Golf, VW Beetle and Allegro in ascending order, are wider, the Allegro with a rear track of 1391 mm is 151 mm wider.

As the diameter of the driven pulley is approximately the same as the wheel diameter, the side plate of the rear suspension arm was extended to protect the belt and pulley and form part of the belt guard.

The use of the Uniroyal Powergrip HTD Toothed Belts, or any other toothed belt, meant that the alignment of the driver and driven-pulleys was critical, as too much misalignment would cause the belt to run off. The object then was to design the rear suspension arm so that the angular misalignment between the shaft of the electric motor and the rear wheel stub axle, was, under all load conditions, within the tolerance allowable for the toothed belt drive. Finite Element Analysis provides a means of determining this misalignment.

The author had used the Finite Element Program PAFEC 70+ previously but, in collaboration with Dr. J. Astley of the University, it was decided that the more sophisticated PAFEC 75 would be more suitable for this task. It was also intended to analyse the chassis structure using Finite Element Analysis. PAFEC 75 has the facility to prepare element data rapidly, using Pafblocks to analyse anisotropic materials and reinforced plastics, and has other advantages over PAFEC 70+. The program PAFEC 75, although available, had not been run on the B6700 at the University of Canterbury before. The debugging process took a considerable amount of time and eventually precluded the analysis of the chassis using Finite Element Methods.

Figure 4.1 illustrates the output from phase 3 of the PAFEC 75 program, showing a plot of the input data, produced to confirm that the geometrical data has been inputted correctly. Figure 4.2 illustrates the output from phase 8 showing a plot of the displaced shape of the suspension arm. Appendix 4.2 gives a brief description of the phases of the PAFEC 75

Figure 4.1. Output from Phase Three.
Drawing of input data to visually confirm
data has been inputed correctly.

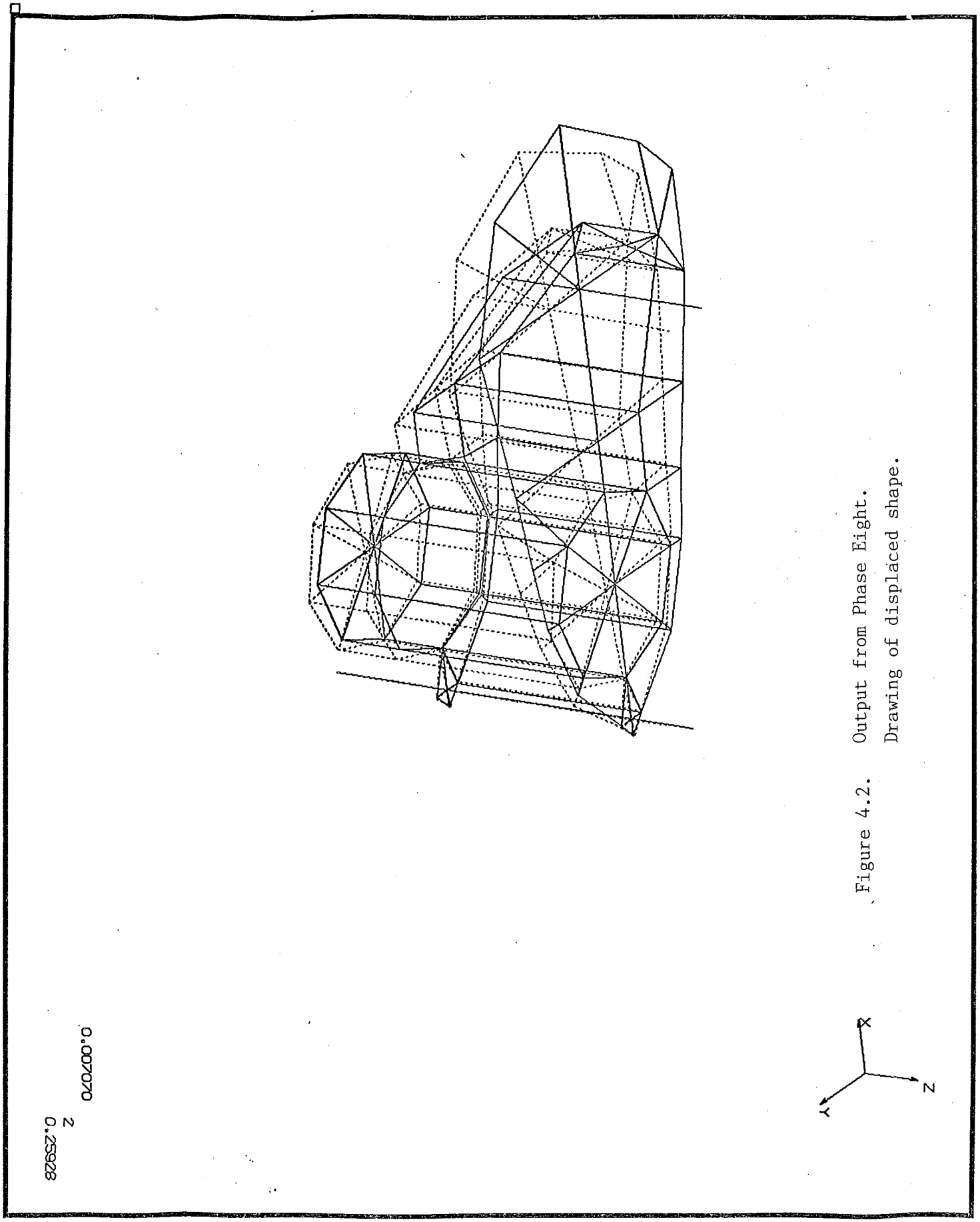


Figure 4.2. Output from Phase Eight.
Drawing of displaced shape.

program, summarizes how the data is prepared and lists the input data used.

Three load cases are considered, the actual forces for which are detailed in Appendix 4.2. Load case 1 is the static 1 g condition. Case 2 is an effective 4.5 g vertical wheel loading. It is agreed that between 2.4 g and 3 g is a safe total wheel load (static plus dynamic overload) (Andrew and Whittaker, 1969 ; Sharman, 1976), but a 50% overload is considered to design purposes to account for fatigue and provide durability, (Costin and Phipps, 1961 ; Skattum, Harris and Howell, 1975); hence a 4.5 g loading. Load case 3, allows for a 3 g vertical wheel loading, combined with a 2 g lateral loading and a 2 g braking load which will cover cornering and braking on a rough road (Skattum, Harris and Howell, 1975).

From the displacement/rotation output, phase 7, illustrated in Appendix 4.2, the misalignment of the driver and driven pulleys was found to be an acceptable 1.7 degrees for load case 2. The subsequent use of chain drive meant that this misalignment was not so critical. The portion of the arm rearward and nominally above the stub axle to protect the belt and pulley was no longer as important, and could be removed to reduce unsprung weight as discussed in Appendix 4.2.

CHAPTER 5

CHASSIS LAYOUT AND DESIGN

5.1 LAYOUT

The general layout, Figure 5.1.1, features a backbone chassis with eight batteries located between the front wheels and six beneath each of the front seats. The rear suspension trailing arms (Chapters 3 and 4) incorporating the electric motors are located beneath the rearward facing rear passengers. The front suspension is a conventional double wishbone suspension and incorporates anti-dive geometry (Chapter 3). The inverters are located under the rear seat between the trailing arm suspension units.

By considering the plan area of the car and comparing it with the combined plan area of wheels and suspension, passengers, batteries, motors and inverters it is apparent that some overlap is required. Having the front passengers sitting above six batteries each and the rear passengers above the rear suspension motors and inverter provides the solution. With all four passengers facing forward, it is normally necessary for the feet of the rear passengers to extend beneath the rear portion of the front seat, however, with this space occupied by batteries, other things being equal, the wheelbase needs to be increased. For safety reasons the batteries should not extend significantly beyond the front line of the front tyres. The front interconnecting torsion bar must fit between the rear of the front batteries and the front passengers, producing a further increase in wheelbase.

Different orientations, layouts and seating arrangements were then postulated, drawn and analysed. Although it is commonly found (Grylls), that the conventional seating arrangement takes up the least space, in this instance it was found that a more compact car could be designed with the two rear passengers facing backwards. This layout was adopted, with the rear seats nominally designed for a 50% male. It was subsequently reported that a proposed (Guess, Nial and Pocobello, 1977) and subsequently built (Shacket, 1979), General Electric Current Technology Electric Vehicle also utilized rear passengers facing backwards.

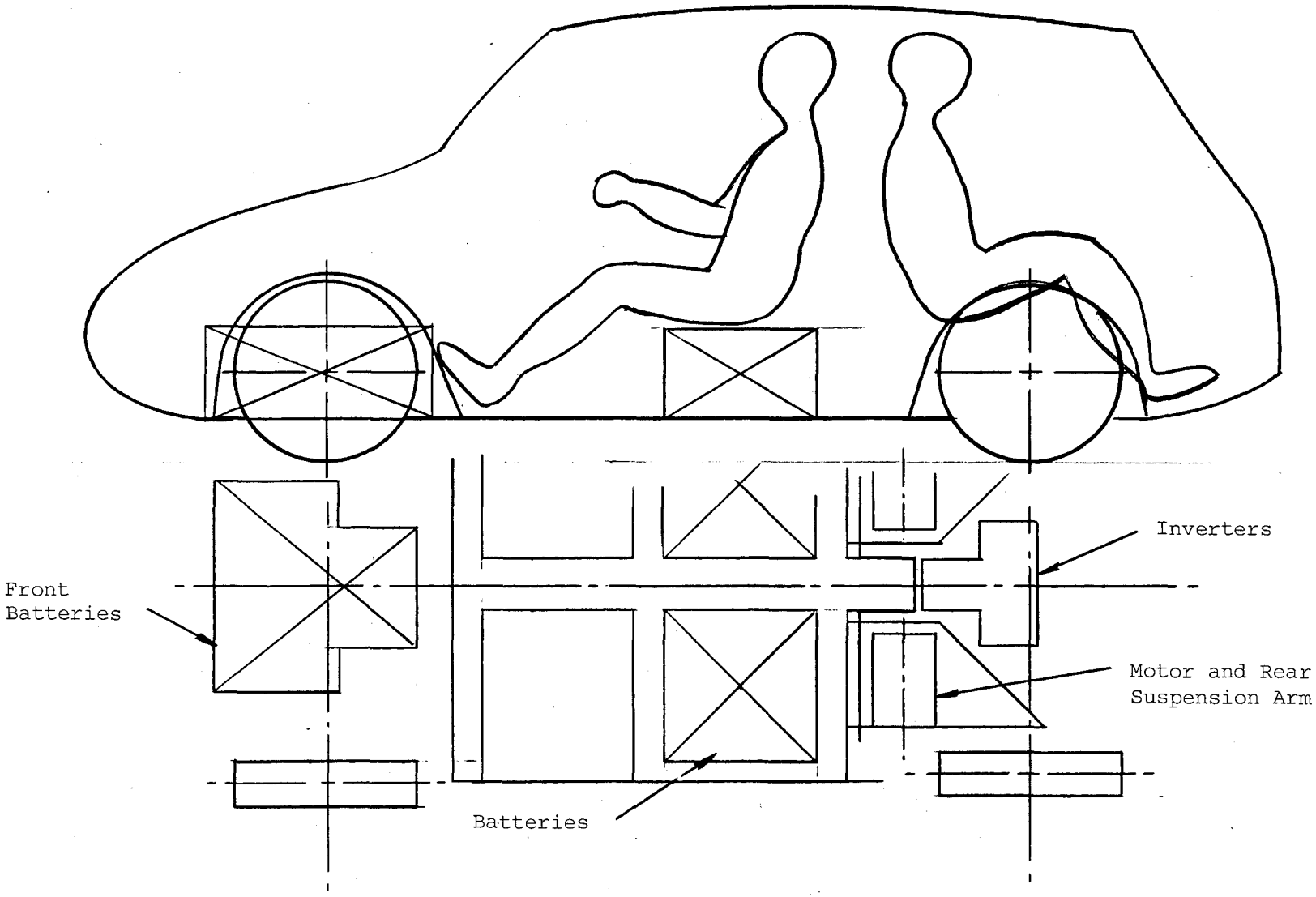


Figure 5.1.1. General Layout.

The backbone chassis is I-shaped in planview, and while the rear suspension is mounted directly to, and behind the rear cross-piece, with side batteries in front of the rear cross-piece, the front suspension and front batteries were to be cantilevered off the front cross-piece. The front batteries and suspension would be supported using a sub-assembly, incorporating progressive crushing and designed to deflect the batteries downward and the main chassis upward in a major front collision, thus preventing battery intrusion into the passenger compartment and protecting passengers from acid. Energy is absorbed in stopping the 131 kg eight battery pack at the front of the car, by the progressive crushing mechanism and by the work done in lifting or deflecting the main chassis structure upward. SAE 740040 gives information on the axial impact crushing (as opposed to buckling) of closed hat structures when loaded axially. The size and thickness is chosen so as to crush at particular impact conditions. The use of the A40 Farina (section 5.3) body meant that the energy absorbing front sub-assembly could not be incorporated, for although the complete floor structure was removed from the Farina body and replaced with a grp-foam sandwich structure, the existing front wheelarch structure which incorporated the suspension mounts and original engine mounts was not changed.

5.2 METHODS OF CONSTRUCTION

For vehicles in limited volume production and with the layout as described in section 5.1, two of the more obvious methods of construction are,

(1) a steel backbone chassis with suspension components attached thereto with a separate glass reinforced plastic (grp) body, and

(2) a complete grp-foam sandwich structure combining the function of body and chassis. The final choice of which may well depend on the design and the production facilities available when manufacture proceeds.

For a complete car a torsional stiffness of 6780 Nm/degree (±50%) and a deflection of 1.3 mm (±50%) under a centrally applied load of 6.67 kN is considered acceptable (Boden). Over a wheelbase of 2.23 m, for the Electric Town Car, this implies a torsional rigidity of 15119 Nm^2/degree (±50%). The steel I-shaped backbone chassis is readily able to be analysed using normal solid mechanics, (Benham and Warnock, 1973) where for example a steel backbone of outside dimensions 300 mm x 150 mm and a thickness of 0.914 mm has a torsional rigidity of 10933 Nm^2/degree, without

bodywork. The steel-backbone would take the suspension loads directly, leaving the moulded grp body structure to support the batteries, inverters, passengers etc. The battery loads are such that extensive use of grp - foam - sandwich would be required. This body could be rubber mounted to the backbone chassis thus reducing the transmission of road noise and the like. The complete mechanical system, comprising chassis, front suspension, rear drive-unit suspension, with appropriate interconnection could be built up and then mated to the body, a practical arrangement for limited volume production.

The steel backbone would normally be chosen because of the combined effects of heat, vibration and concentrated loading due to the internal combustion engine on a grp structure. The Electric Town Car has no such combination of problems. The effect of concentrated suspension loads can be reduced by using a wide load base, by feeding large inertia loads directly into the suspension and by appropriate detail design and development.

While the tensile strength of certain grp composites is comparable to, and the specific strength in excess of steel and aluminium, grp composites have a modulus of elasticity approximately an order of magnitude below that of aluminium and steel (Warring, 1971). The properties of grp composites are dependent on the choice of resin and the type and form of glass reinforcement, and are well documented (for example Parkyn, 1970). Further, the reproducibility of the properties is also dependent on moulding and layup techniques and the glass/resin content, so reduced mechanical properties should be used (Oleesky and Mohr, 1964). For a more technical treatise on the subject of reinforced plastics MIL-HDBK-17A (1971) should be consulted.

In order to overcome the problems associated with the relatively low elastic modulus, it is necessary to incorporate curvature or stiffening ribs into the mouldings or to use sandwich construction (Parkin). The use of sandwich construction and in particular sandwich construction using a foam core is of interest. The grp facings provide the strength and the foam, acting rather like the web on an I-beam, produces the separation and thus the mechanism for the increased rigidity. The grp facings are considerably thinner than in the non-sandwich construction and therefore are susceptible to buckling or wrinkling if unsupported. The foam core must therefore support the compression facing against wrinkling and buckling, allowing it to develop its full compressive strength.

Of the three most common rigid plastic foams available, polyurethane is the most practical for this application, pvc the best structurally and polystyrene the least suitable. Polystyrenes are somewhat limited to mass-production techniques, they are not adaptable to in-situ application, are chemically attacked by polyester resins (though not by epoxies) and are susceptible to static creep, (Ferrigno, 1967). A cross-linked, fluorocarbon-gas-filled PVC foam was developed by a French Company, Kleber Colombus S.A. and first used for military purposes in 1950. It exhibits higher strength/weight values than most rigid polyurethane foams and is ideally suited for a foam core in a grp-foam sandwich panel, however it is considerably more expensive than polyurethane rigid foams and cannot be foamed in-situ, (Harper, 1975). Rigid Polyurethane foam has been used in a number of diverse applications, its choice in most cases being dependent on four characteristics that distinguish it from similar materials, (Backus and Geinhardt, 1973).

(1) Urethane foams are easily prepared from liquid raw materials.

(2) Properties can be changed by appropriate selection of raw materials and foam density.

(3) Thermal conductivity is the lowest among the common insulating materials.

(4) Mechanical strength is high at low density.

An Experimental Plastic Car, developed by Bayer in Düsseldorf (October 1967) demonstrated the versatility of urethane foam. The car used a load-beaming base unit, that completely replaced the conventional chassis. The base unit and front fenders are a sandwich construction with outer skins or shells of epoxy-fibreglass and the cavity filled with rigid urethane foam (96 kg/m^3). The rear fenders are rigid urethane foam-core sandwich construction, with thermoplastic facings. The bonnet and boot are made from a special high density rigid urethane foam using a technique called Reaction Injection Moulding (RIM). The Citroen-Mahari (a specialized off-road vehicle) used a self-supporting floor assembly consisting of two shells of glass reinforced polyester, with urethane foam injected between them, resulting in a very stiff sandwich structure, (Hablitzel, 1973). Under conditions of extreme stress, a 64 kg/m^3 urethane has been found to provide excellent durability, the foam has been used as a core for helicopter blades, upon which the leading edge spar and trailing edge skin are bonded, (Ferrigno, 1967).

It was proposed that the first vehicle to this electric town car design be made by glassing over a foam buck, rather like making a surfboard. That is, the chassis would be made into the final shape in foam and the glass reinforced plastic applied to that. For any subsequent but identical cars, the original would be used to make a pair of moulds. The skins produced from these moulds would be bolted or bonded together and the foam moulded between them in-situ, producing the sandwich structure. This method of construction restricts the foam to the thermosetting polyurethane foam, which forms a strong adhesive bond with a wide range of subtrates including metals, glass and wood (Beadle) and with both polyester and epoxy grp laminates (Giles) provided they are free of grease. It would be necessary to use epoxy resin for the first vehicle as this would provide the necessary adhesion to the foam, in the first instance, whereas polyester resin would not (Beale, 1962).

The use of reinforced plastic-polyurethane core sandwich construction is extremely versatile, whether used as a unitary body/chassis construction or in combination with a steel backbone chassis. The mechanical properties of polyurethane foam are approximately proportional to the foam density (Ferrigno, 1967), so that the shear strength and modulus (bending) and compressive strength (buckling and wrinkling of the facings) can be tailored to the particular application. The resins can be chosen for chemical resistance and/or fire retardation properties in the polyesters and can be used adjacent to each other in the same lay-up. Epoxies are self-extinguishing and offer excellent chemical resistance, they are stronger though more expensive than the polyesters. The thickness of the facings can be varied to suit stress levels and can be locally thickened at points of load application or inserts can be utilized. The choice and orientation of rovings and cloth, (chopped strand mat is not normally used for such structural applications), can produce a facing that is orthotropic or quasi-isotropic or anywhere in between, so that the directional strength of the facings can be aligned with the principal stress directions. While glass fibres are the most common reinforcement, aramid fibres such as Kevlar 49 can also be considered for selected reinforcement in this application, (Anon, 1974; Jarrell, 1977; Clements and Chiao, 1977).

In order to establish accurately the principal stress directions in, and the rigidity of, the unitary chassis/body structure, it was planned to analyse the structure using the finite element method. Although this was not done (section 4.2), literature research had been completed to determine the location of different types (beam, plate, etc.) of

elements to produce the most computer cost effective and yet realistic results (Horvath, 1975; Hedges, Norville and Gurdogen, 1971; Radaj, Zunmer and Geissler, 1970 ; Hessel and Lammers 1971; Alaylioglu and Ali, 1977), and the type of analysis, static and dynamic, (Peterson, 1971 ; Ali, Hedges and Mills, 1971).

5.3 ADAPTION OF A40 FARINA BODYWORK

The Electric Town Car as built utilizes conventional steel bodywork with a grp-foam sandwich floor and backbone structure. The car body is based on an Austin A40 Farina with extensive modifications so that only the side panels, side doors, side windows, windscreen, roof and front suspension support structure remain. The decision to utilize an existing body to provide the outer skin, doors and windscreen was made as an economy and time saving measure. The Farina was chosen because it had a wheelbase consistent with the Electric Town Car requirements and it was known to have a good aerodynamic shape, (White, 1967).

The modifications to the body included removing the front fenders above an existing longitudinal horizontal seam and welding in a replacement internal panel, replacing and lowering the front headlights to the minimum regulation 600 mm, attaching a grp nose to the front of the car and an appropriate grp bonnet, as well as fabricating a large gas strut assisted lift-up rear door. The floor was completely removed, some steel structure was added for mounting suspension components and to provide an interface for the grp-foam sandwich floor. These structural modifications are detailed on the drawings in Appendix 6.1.

5.4 GRP FOAM SANDWICH ANALYSIS AND CONSTRUCTION

The use of the A40 Farina bodywork provides a considerable amount of stiffness both in bending and in torsion to the extent that the grp. sandwich structure will not be fully tested. The grp sandwich floor panels were designed to withstand the 3 g battery inertia loading, and the backbone checked for deflection. The floor panel was made by foaming polyurethane rigid foam in-situ between two glass reinforced polyester panels thus simulating the production situation, while the front and rear crosspieces and the battery surround were made by glassing over rigid polyurethane foam using epoxy resin. Because of creep characteristics, at a sustained

loading generally only 50% of the tensile strength of thermoplastics and 70% in the case of thermosetting materials is usable (Hablitzel, 1973). Experiments on the creep of rigid urethane foam in shear (Hartsock, 1967), indicate that the creep deflection tends to level off after a period, dependent on the foam density and level of stress. Experiments indicate that foamed in-situ urethane foam is anisotropic, though it is far from orthotropic in the same sense that a honeycomb is orthotropic, (Sayigh, 1966). The strength properties across the rise are generally 60% to 80% of those in the direction of foam rise (Beadle). It has been suggested (Hartsock, 1966) that the shear strength (though not shear modulus) of in-situ foam mouldings could be as low as one-third that of free rise foams of similar density if care is not taken to minimise flow during foaming.

The side batteries and five of the front batteries are supported directly by the floor panels. The side batteries are the worst case, with a uniformly distributed loading of q = 4.45 kPa in the static case, on a panel size of 472 x 420 mm. Using ECK-12 glass woven roving, 363 gm/m^3, nominal thickness 0.46 mm with a warp x weft count (per inch) of 10 x 9, and orientating this in three layers at $0^o/45^o/90^o$ will produce a quasi-isotropic grp facing, which for design purposes may be considered as isotropic. While the foam is anisotropic, it is at least continuous (honeycomb cores for example are not) and it is suggested (Allen, 1969) that it be considered isotropic for design purposes. The problem simplifies considerably to one of an isotropic face and an isotropic core. Further, if the facings are considered as membranes, the central deflection of a panel of any aspect ratio can be obtained (Erickson, 1950) using the formula

$$W_{max} = \frac{qa^4\lambda_f}{IE_f} \quad \alpha_1(1 + S\alpha_2)$$

where a = 0.42 m (the short side) and b = 0.472 m and for a/b = 0.89, α_1 = 0.0051 and α_2 = 1.65. Where ν is Poissons ratio, $\lambda_f = 1-\nu^2 = 0.947$.

$$S = \frac{ct\pi^2E_f}{2a^2\lambda_f G_c} = 1.59$$

Using the appropriate reduced mechanical properties for ECK-12, σ_f = 152 MPa and E_f = 14 GPa in polyester resin, and for 96 kg/m^3 polyurethane G_c = 8 MPa and τ_c = 706 kPa (not reduced). For an overall panel thickness of 25 mm, and individual facing thicknesses t = 1.38 mm each, the core thickness C = 22.24 mm, then

$$I = \frac{t_f}{2} (C + t_f)^2 = 3.85\ (10)^{-7}$$

and maximum deflection is 0.45 mm (w/a = 1/935) in the static condition and maximum deflection is 1.35 mm (w/a = 1/312) in the 3 g inertia load condition.

The maximum bending moment is $M = \beta.q.a^2$ (Oleesky and Mohr, 1964), where for a/b = 0.89, $\beta_a = 0.0570$, $\beta_b = 0.0493$ and $\gamma = 0.4417$, so that M = 134.3 Nm (using 3 g inertia load). The maximum facing stress $\sigma_f = M/t_c.t_f$ where $t_c = C = 22.24$ mm and $t_f = t = 1.38$ mm is 4.38 MPa. The maximum core shear stress $\tau_c = 2\gamma qa/(t+t_c)$, with t = 25 mm is 105 kPa. The sandwich panel is therefore satisfactory, and with a static core shear stress of 35 kPa, creep will not be a problem, although the static deflection 0.45 mm will be expected to increase, perhaps doubling to approximately 1 mm (Ferrigno, 1967).

The deflection of the central backbone tunnel made up of the same 25 mm sandwich panel as above, was found to be 0.08 mm under the 6.67 kN centrally applied load as discussed in section 5.2.

The panels as built, in error, by the grp subcontractor had chopped strand mat between the woven rovings which effectively doubled the thickness of the facings. This will not double the strength or stiffness, as chopped strand mat is not as strong or stiff as woven rovings. An increase in stiffness of 50% might be expected, (Waring, 1971).

CHAPTER 6

CONCLUSION

The proposed Electric Town Car is being built. It has been possible to accommodate four adults in this small car by having the rear passengers facing rearwards, above the rear suspension drive unit and each of the front passengers above six batteries.

Low rolling resistance and low aerodynamic drag has been achieved by using a simple but efficient drive system utilizing a single reduction chain drive to each of the rear wheels and by aerodynamic modifications to improve the drag coefficient of the bodywork, which was based on an A40 Farina bodyshell, itself recognized to have low drag coefficient. Special low energy-loss tyres are used.

The requirements of comfortable vehicle ride are met by the use of a torsion-bar interconnected suspension, allowing the pitch frequency to be reduced to nominally 42 c.p.m., and the bounce frequency maintained at nominally 75 c.p.m. The suspension geometry has been chosen to maintain body pitch and roll angles at acceptable levels during braking and cornering. Self-levelling is required to compensate for static load variation.

An electric motor and chain drive is incorporated in each rear suspension arm. Toothed belt drive was originally proposed, and the rear arm was designed to minimise deflections during severe loading, to avoid belt run-off, resulting in suspension arms of excessive weight.

Two methods of body/chassis construction are proposed and discussed. The use of a steel backbone chassis and fibreglass body will require less development time before manufacture can commence, whereas the unitary fibre-glass-foam sandwich construction method is likely to be more economical to manufacture, as the separate chassis is eliminated but requires more design and development time. The electric town car as built incorporates features from both methods of construction.

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APPENDIX 2.4.1

INTERCONNECTED SUSPENSION PROGRAM

DATA INPUT AND ANSWER CODE HANDBOOK

Data Input:

All data input is stored in a single array "Data List" labelled DL. The input units are SI. The last data card must be blank (i.e. the ninth data card is blank).

Answers:

All calculation results or answers are stored in a single array labelled "ANS".

First Data Card Format 8I5

Control Card

Columns	5	10	15	20	25	30	35	40
	ITF	ITR	IAL	ITA	IVG	LBA	ITA	
DL	(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)

ITF - Front torsion bar
- 0 - no date/calculations.
- 1 - circular torsion bar.
- 2 - rectangular torsion bar.

ITR - Rear torsion bar
- 0 - no data/calculations.
- 1 - circular torsion bar.
- 2 - rectangular torsion bar.

IAL - Arm length data
- 0 - no data/calculations.
- 1 - calculate.

IAD - Initial arm data
- 0 - no data/calculations.
- 1 - calculate.

IVG - Vehicle geometry
- 0 - no data/calculations
- 1 - calculate.

LBA - Load/brake/acceleration
- 0 - no data/calculations
- 1 - calculate.

ITA - Auxiliary t'bar
- 0 - no data/calculations.
- 1 - circular torsion bar.
- 2 - rectangular torsion bar.

DL (8) for future use.

Second Data Card Format 8F 10.4

Front Torsion Bar ITF = DL(1)

ITF = 0 Blank card.

ITF = 1 Circular torsion bar.

Columns	10	20	30	40	50	60	70	80
	G	L	DIAM					
DL	(9)	(10)	(11)	(12)	(13)	(14)	(15)	(16)

ITF = 2

Columns	10	20	30	40	50	60	70	80
	G	L	a	b	η_2	η_3		
DL	(9)	(10)	(11)	(12)	(13)	(14)	(15)	(16)

G = Shear modulus N/m^2

L = Length of torsion bar m

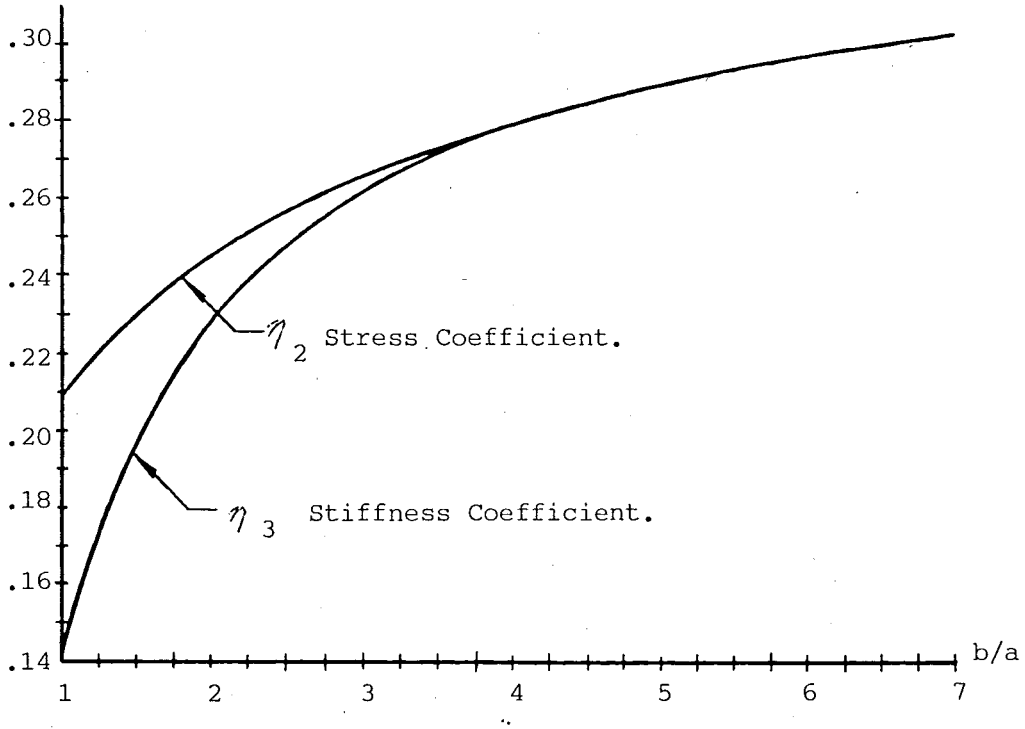
DIAM = Diameter of circular section m

a = 'Short' side of rectangular section m

b = 'Long' side of rectangular section m

η_2 = Saint Venant's stress coefficient.

η_3 = Saint Venant's stiffness coefficient.



Saint Venant's Coefficients for Rectangular Bar in Torsion.

Third Data Card Format 8 F 10.4

Rear Torsion Bar ITR = DL(2)

ITR = 0 Blank card.

ITR = 1 Circular Torsion Bar.

Columns	10	20	30	40	50	60	70	80
	G	L	DIAM					
DL	(17)	(18)	(19)	(20)	(21)	(22)	(23)	(24)

ITR = 2 Rectangular Torsion Bar.

Columns	10	20	30	40	50	60	70	80
	G	L	a	b	η_2	η_3		
DL	(17)	(18)	(19)	(20)	(21)	(22)	(23)	(24)

G = Shear Modulus N/m^2

L = Length of torsion bar m

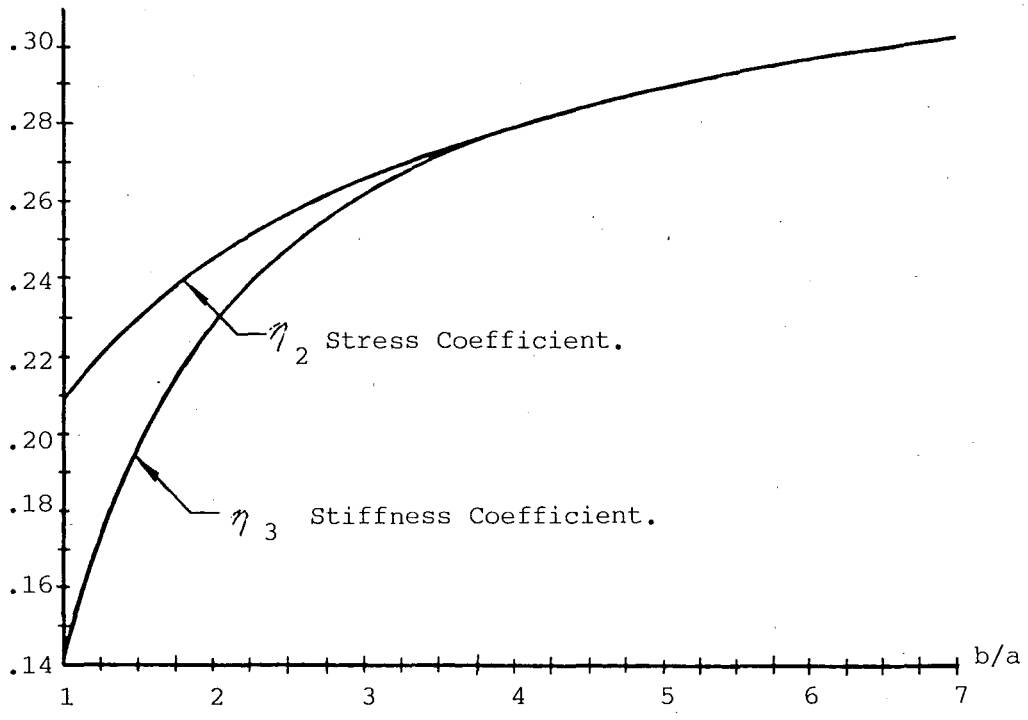
DIAM = Diameter of circular section m

a = 'Short' side of rectangular section m

b = 'Long' side of rectangular section m

η_2 = Saint Venant's Stress Coefficient.

η_3 = Saint Venant's Stiffness Coefficient



Saint Venant's Coefficients for Rectangular Bar in Torsion.

Fourth Data Card Format 8 F 10.4

Arm Length Data IAL = DL(3)

IAL = o Blank card.

IAL = 1 Actual length of arms.

Columns	10	20	30	40	50	60	70	80
	AL1	AL2	AL3	AL4	AL5	AL6	L3	
DL	(25)	(26)	(27)	(28)	(29)	(30)	(31)	(32)

AL1 = arm length 1 = Front wheel to tbar CL

AL2 = arm length 2 = Rear wheel to tbar CL.

AL3 = arm length 3 = Front tbar CL to cable joint.

AL4 = arm length 4 = Rear tbar CL to cable joint.

AL5 = arm length 5 = Front to rear tbar centre distance.

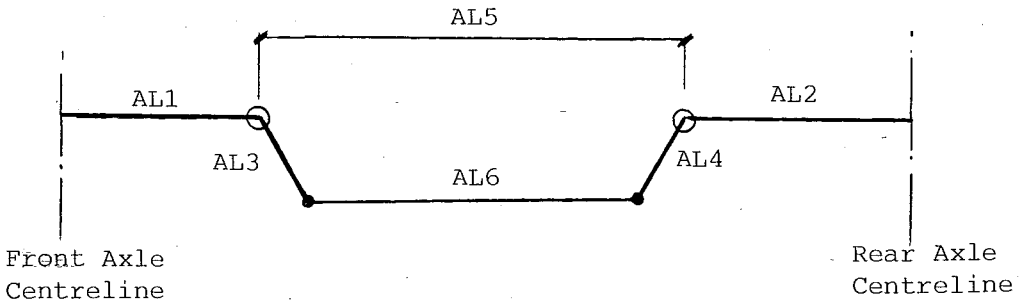
AL6 = arm length 6 = cable length.

L3 = Effective length of front pitch arm.

Note (1) As program is currently written, L3 must be specified and AL3, AL4 and AL6 are calculated, i.e. L3 is specified and AL3, AL4 and AL6 are inputted as zero.

(2) All lengths in metres.

(3) "t bar CL "means" torsion bar centreline".



Fifth Data Card Format 8F 10.4

Initial Angle Data IAD = DL(4)

IAD = 0 Blank card.

IAD = 1 Initial angles.

Columns	10	20	30	40	50	60	70	80
	A1	A2	A3	A4	A5	A6		
DL	(33)	(34)	(35)	(36)	(37)	(38)	(39)	(40)

A1 = Angle of front arm (AL1) from longitudinal axis. Clockwise positive.

A2 = Angle of rear arm (AL2) from longitudinal axis. Clockwise positive.

A3 = Acute angle of arm (AL3) from longitudinal axis.

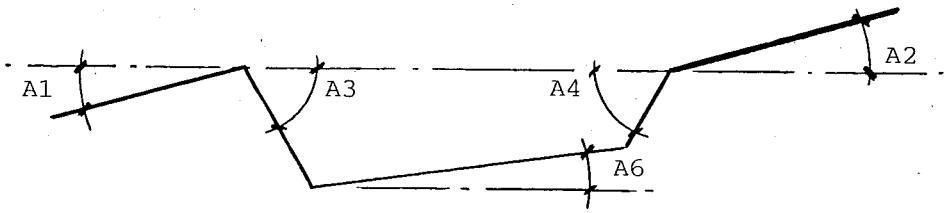
A4 = Acute angle of arm (AL4) from longitudinal axis.

A5 = 0 (Torsion bars same height).

A6 = Angle of cable from longitudinal axis.

Note (1) As program is currently written specify A3 and A4 and program calculates A6.

(2) All angles in radians.



Sixth Data Card Format 8F 10.4

Vehicle Geometry IVG = DL:(5)

IVG = 0 Blank card.

IVG = 1 Vehicle geometry

Columns	10	20	30	40	50	60	70	80
	WB	A	B	H	M	K		
DL	(41)	(42)	(43)	(44)	(45)	(46)	(47)	(48)

WB = Wheelbase. metres.

A = Distance of C of G from front axle as a ratio of the wheelbase ($0 < A < 1$).

B = Distance of C of G from rear axle as a ratio of the wheelbase ($0 < B < 1$).

H = Height of the C of G as a ratio of the wheelbase.

M = Vehicle mass in kg.

K = Radius of gyration as a ratio of the wheelbase.

Note: $A + B = 1$ by definition. If $A + B \neq 1$ the program will abort.

Seventh Data Card — Format 8F 10.4

Load - Brake - Acceleration — LBA = DL(6)

LBA = 0 — Blank card.

LBA = 1 — Load-Brake-Acceleration data.

Columns	10	20	30	40	50	60	70	80
	DFL	MFL	DRL	MRL	DB	MB	DA	MA
DL	(49)	(50)	(51)	(52)	(53)	(54)	(55)	(56)

DFL = Delta front load, kg.

MFL = Maximum front load, kg. (i.e. apply a variable additional static load at the front up to MFL in steps of DFL).

DRL = Delta rear load, kg.

MRL = Maximum rear load, kg. (i.e. apply a variable additional static load at the rear up to a MRL in steps of DRL).

DB = Delta braking acceleration, g's.

MB = Maximum braking acceleration, g's. (i.e. apply variable braking loads up to MB in steps of DB).

DA = Delta acceleration force, g's.

MA = Maximum acceleration force, g's. (i.e. apply variable acceleration loads up to MA in steps of DA).

Note — The purpose of applying these additional loads is to examine the pitch variation due to the application of various load inputs.

Eighth Data Card Format 8F 10.4

Auxiliary Torsion Bar ITA = DL(7)

ITA = 0 Blank card.

ITA = 1 Circular torsion bar.

Columns	10	20	30	40	50	60	70	80
	G	L	DIAM					
DL	(57)	(58)	(59)	(60)	(61)	(62)	(63)	(64)

ITA = 2 Rectangular torsion bar.

Columns	10	20	30	40	50	60	70	80
	G	L	a	b	η_2	η_3		
DL	(57)	(58)	(59)	(60)	(61)	(62)	(63)	(64)

G = Shear Modulus. N/m^2

L = Length of torsion bar, m.

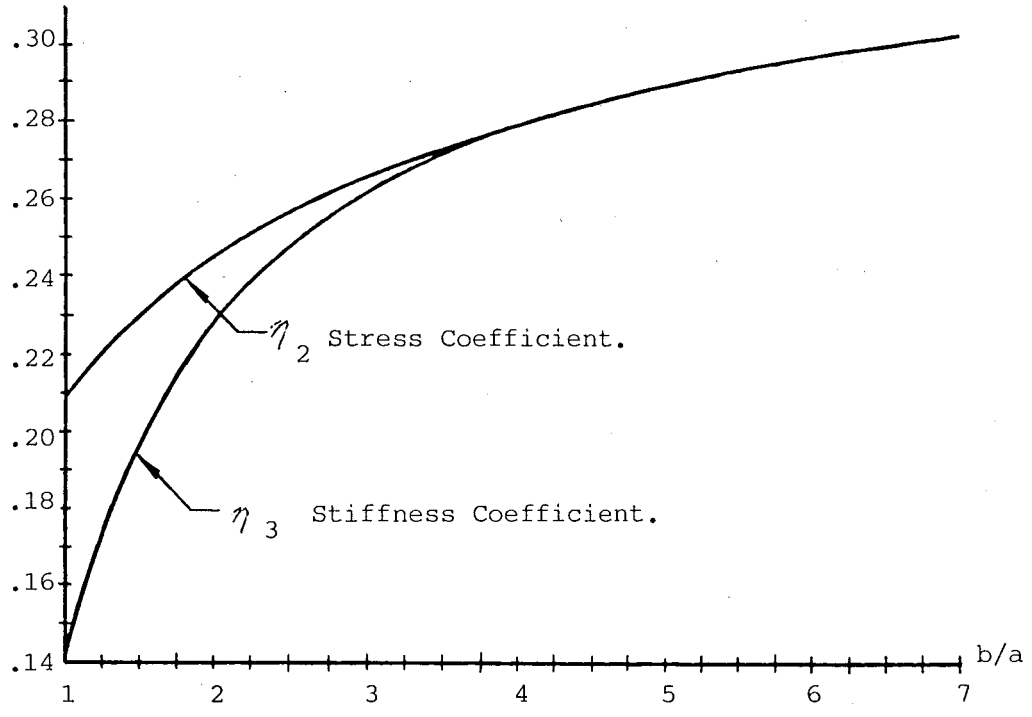
DIAM = Diameter of circular section, m.

a = 'Short' side of rectangular section, m.

b = 'Long' side of rectangular section.

η_2 = Saint Venant's Stress Coefficient.

η_3 = Saint Venant's Stress Coefficient.



Saint Venant's Coefficients for Rectangular Bar in Torsion.

CALCULATED RESULTS

(Array ANS. abbreviation for answers).

ANS(1) = Front torsion bar springrate, kNm/radian. = C_1

ANS(2) = Front torsion bar stressrate, MPa/radian.

ANS(3) = C_2 = Rear torsion bar springrate, kN m/radian.

ANS(4) = Rear torsion bar stressrate, MPa/radian.

ANS(5) = C_3 = Auxiliary torsion bar springrate, kN m/radian.

ANS(6) = Auxiliary torsion bar stressrate, MPa/radian.

ANS(7) = Initial ratio of length = $\frac{B}{A} \frac{AL1 \cos A1}{AL2 \cos A2} = \frac{\ell_3}{\ell_4}$

ANS(8) = Cable angle A6, radians.

ANS(9) = Angle of front pitch arm A3, radians.

ANS(10) = Angle of rear pitch arm A4, radians.

ANS(11) = Front locked pitch angle on front bump. Unequal pitch arms. (i.e. angle of front pitch arm corresponding to the condition in ANS(12)), radians.

ANS(12) = Rear locked pitch angle on front bump. Unequal pitch arms. (i.e. angle at which the rear pitch arm and the cable are in a straight line), radians.

ANS(13) = Front locked pitch angle on rear bump. Unequal pitch arms. (i.e. angle at which the front pitch arm and the cable are in a straight line), radians.

ANS(14) = Rear locked pitch angle on rear bump. Unequal pitch arms. (i.e. angle of rear pitch arm corresponding to the condition in ANS(13)), radians.

ANS(15) = Angle A1, radians.

ANS(16) = Angle A2, radians.

ANS(17) = Angle θ_1, positive for front bump, radians.

ANS(18) = Angle θ_2, positive for rear bump, radians.

ANS(19) = Angle θ_3, positive for front bump, radians.

ANS(20) = Angle θ_4, positive for rear bump, radians.

ANS(21) = Effective length ℓ_1 = AL1.Cos A1, m.

ANS(22) = Effective length ℓ_2 = AL2.Cos A2, m.

ANS(23) = Effective length ℓ_3 = AL3.Sin (A3-A6), m.

ANS(24) = Effective length ℓ_4 = AL4.Sin(A4 + A6), m.

ANS(25) = Ratio of effective lengths RL34 = ℓ_3/ℓ_4.

ANS(26) = Ratio of angles RA34 = θ_3/θ_4.

ANS(27) = RATIO $= \frac{\ell_3}{\ell_4} \cdot \frac{\theta_4}{\theta_3} = \frac{RL34}{RA34}$

ANS(28) = Cable Tension T, kN.

ANS(29) = Front wheel displacement f_1, m. bump positive (up).

ANS(30) = Rear wheel displacement f_2, m. bump positive (up).

ANS(31) = Front wheel reaction P_1, kN.

ANS(32) = Rear wheel reaction P_2, kN.

ANS(33) = Springrate C_{11} (wheelrate), kN/m.

ANS(34) = Springrate C_{22} (wheelrate), kN/m.

ANS(35) = Springrate C_{12} (wheelrate), kN/m.

ANS(36) = Springrate C_{21} (wheelrate), kN/m.

ANS(37) = Partial bounce frequency ω_z = PFB, Hz.

ANS(38) = Partial pitch frequency ω_α = PFP, Hz.

ANS(39) = Coupling coefficient η_{C1} = CC1.

ANS(40) = Coupling coefficient η_{C2} = CC2.

ANS(41) = Pitch frequency Ω_1 = PF, Hz.

ANS(42) = Bounce frequency Ω_2 = BF, H_z.

ANS(43) = ω_z^2 = PFB**2, $(\text{radian/sec})^2$.

ANS(44) = ω_α^2 = PFP**2, $(\text{radian/sec})^2$

ANS(45) = Ω_1^2 = PF**2, $(\text{radian/sec})^2$.

ANS(46) = Ω_2^2 = BF**2, $(\text{radian/sec})^2$.

ANS(47) = Angle θ_2 at zero wheel loading, radians.

ANS(48) = Angle θ_2 at zero wheel loading, radians.

ANS(49) = f_{02} = static front wheel deflection, m.

ANS(50) = f_{02} = static rear wheel deflection, m.

ANS(51) = G_1 = static front wheel loading, kN.

ANS(52) = G_2 = static rear wheel loading, kN.

ANS(53) = 2*C11 = Total front springrate, kN/m.

ANS(54) = 2*C22 = Total rear springrate, kN/m

ANS(55) = $2*C_{12}$ = Total interconnected springrate, front due to rear, kN/m.

ANS(56) = $2*C_{21}$ = Total interconnected springrate, rear due to front, kN/m.

ANS(57) = P_1/G_1 = Dynamic/Static wheel load.

ANS(58) = P_2/G_2 = Dynamic/Static wheel load.

ANS(59) = θ_1 = Windup angle for torsion bar C_1, radians.

ANS(60) = θ_2 = Windup angle for torsion bar C_2, radians.

ANS(61) = S_1 = Stress for torsion bar C_1, MPa.

ANS(62) = S_2 = Stress for torsion bar C_2, MPa.

ANS(63) = S_3 = Stress for torsion bar C_3, MPa.

APPENDIX 2.4.2

INTERCONNECTED SUSPENSION ANALYSIS, INTERSUSP.

Purpose. To calculate suspension characteristics of the cable interconnected torsion bar suspension as designed and built for the Electric Town Car.

Input* Vehicle spring and physical properties plus initial lever arm length and angles.

Output* Individual torsion bar properties, pitch arm and cable lengths and static wheel deflections.

For different permutations of front and rear bumps, the following are printed:

F1	-	front wheel displacement.
F2	-	rear wheel displacement.
P1	-	front wheel reaction force.
P2	-	rear wheel reaction force.
THT 1	-	angular displacement of A1.
THT 2	-	angular displacement of A2.
THT 3	-	angular displacement of A3.
THT 4	-	angular displacement of A4.
P1/G1	-	ratio dynamic/static force (front).

* ref "Interconnected Suspension Program, Data Input and Answer Code Handbook".

P2/G2	=	ratio dynamic/static force (rear).
RATIO	=	$\ell 3/\ell 4 \cdot \theta 4/\theta 3 = \frac{RL34}{RA34}$.
T	=	cable tension, kN.
ITS	=	number of iterations requires within program.

The following wheelrates,

C_{11}	=	front springrate due to front displacement.
C_{22}	=	rear springrate due to rear displacement.
C_{12}	=	front springrate due to rear displacement.
C_{21}	=	rear springrate due to front displacement.

The following frequencies,

pitch frequency,

bounce frequency,

partial frequency - pitch and bounce,

coupling coefficients.

The following torsion bar properties,

PHI 1 = windup angle of torsion bar 1,

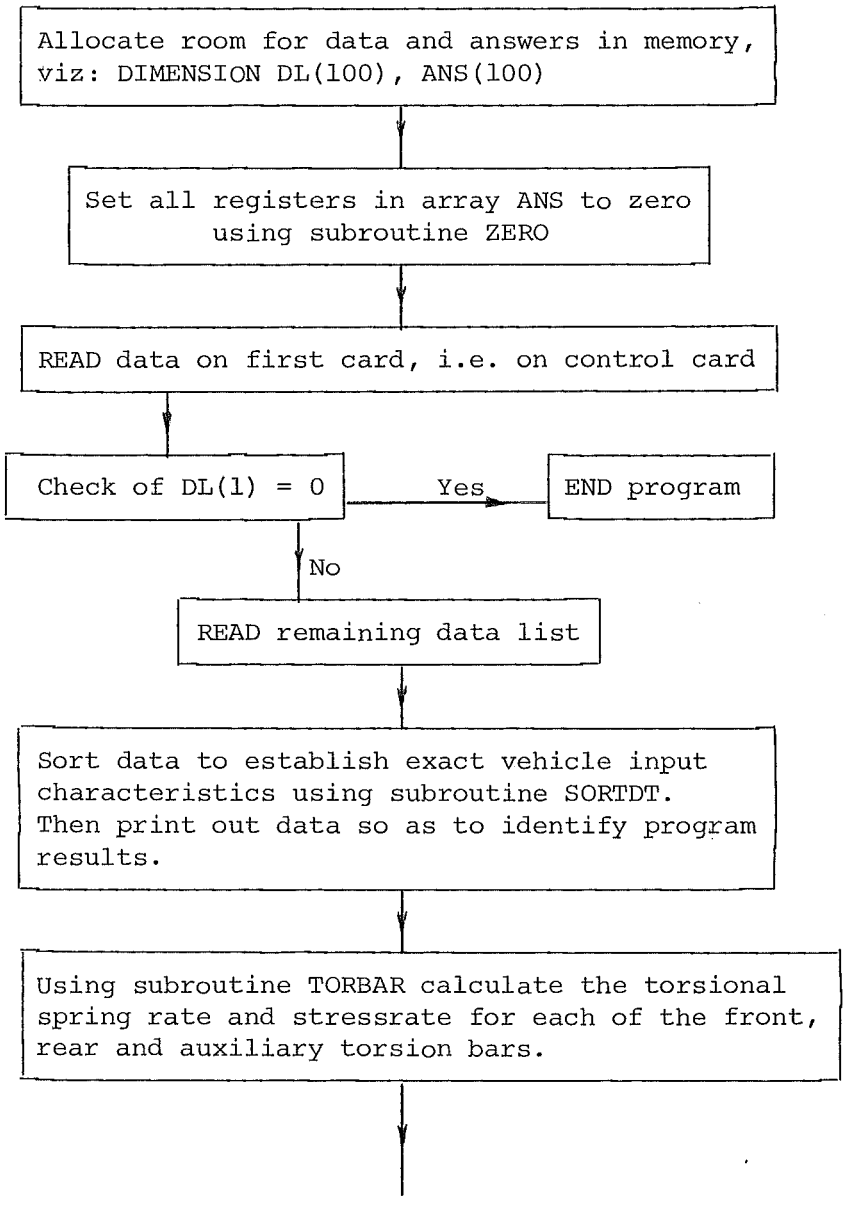
PHI 2 = windup angle of torsion bar 2,

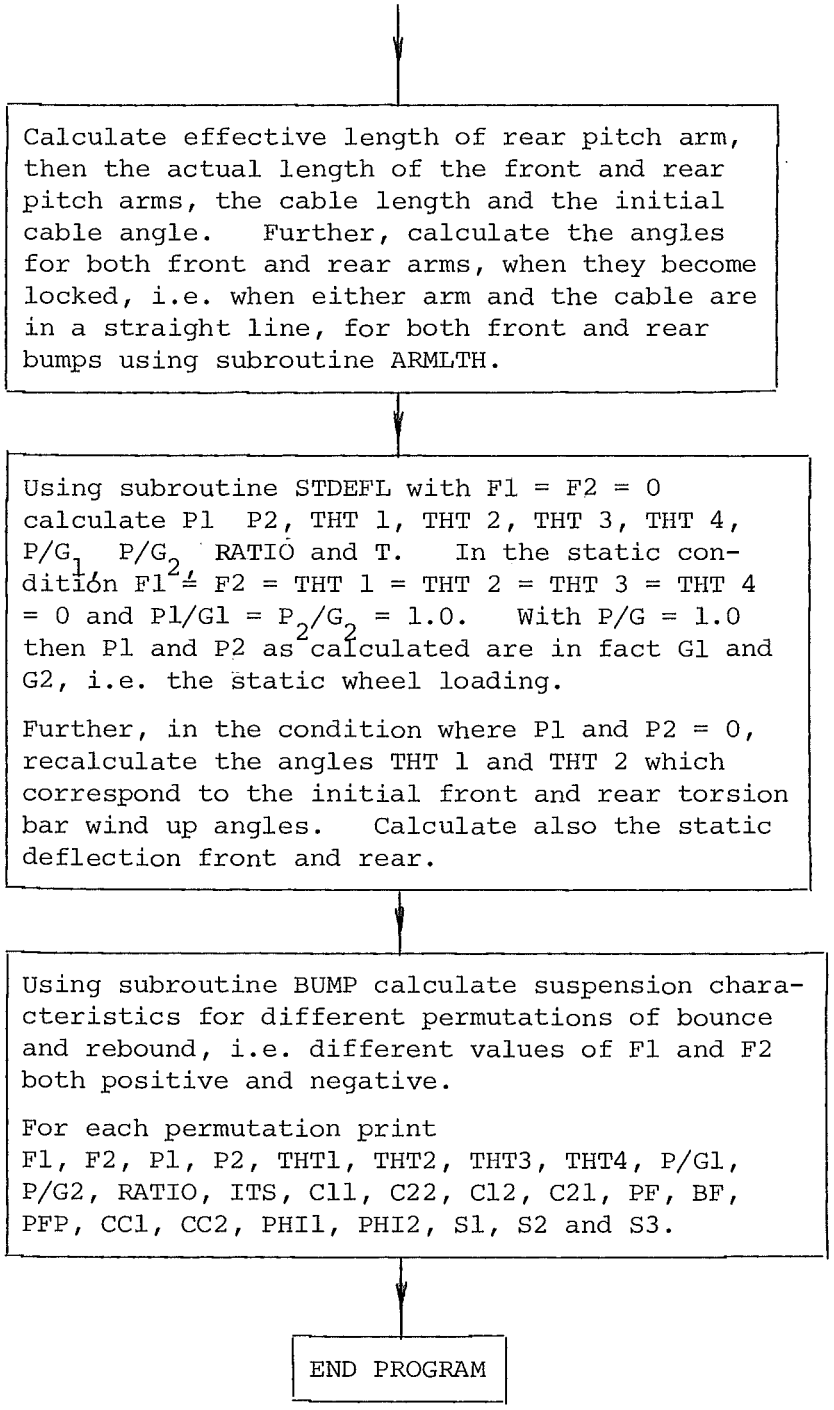
S1 = stress in torsion bar 1,

S2 = stress in torsion bar 2,

S3 = stress in auxiliary torsion bar.

FLOWCHART

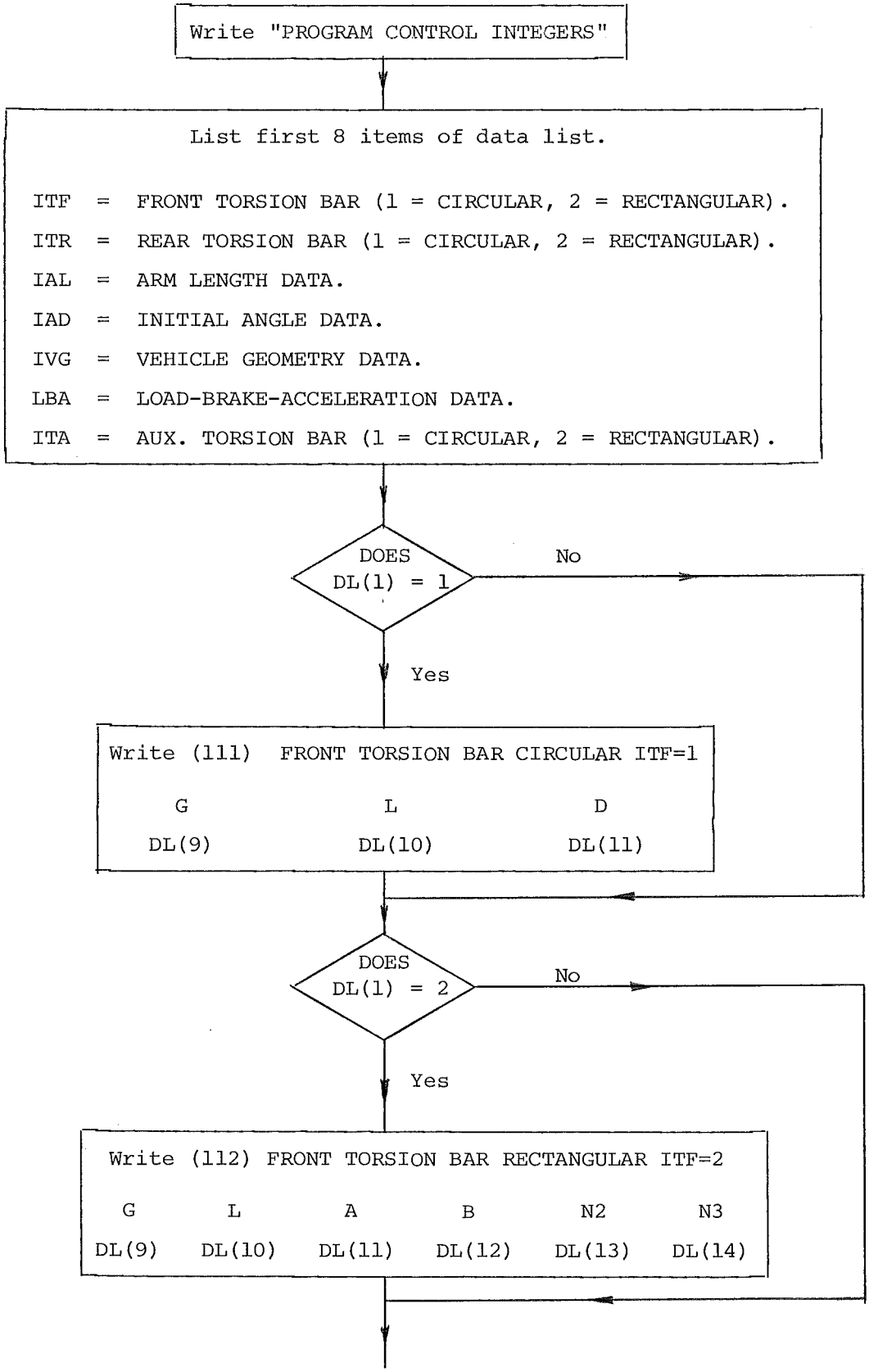


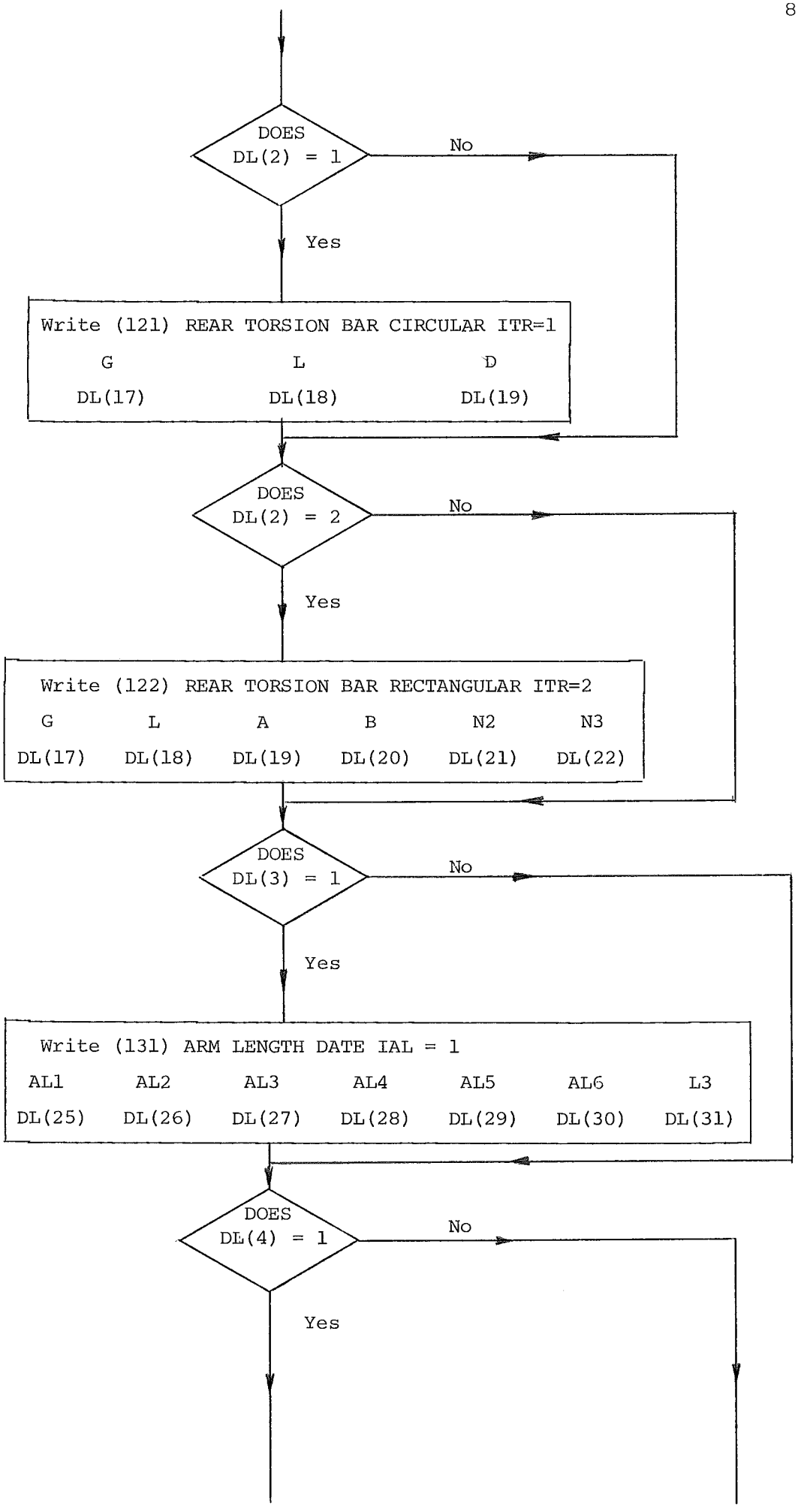

Calculate effective length of rear pitch arm, then the actual length of the front and rear pitch arms, the cable length and the initial cable angle. Further, calculate the angles for both front and rear arms, when they become locked, i.e. when either arm and the cable are in a straight line, for both front and rear bumps using subroutine ARMLTH.
Using subroutine STDEFL with F1 = F2 = 0 calculate P1 P2, THT 1, THT 2, THT 3, THT 4, P/G1 P/G2 RATIO and T. In the static condition F1 = F2 = THT 1 = THT 2 = THT 3 = THT 4 = 0 and P1/G1 = P2/G2 = 1.0. With P/G = 1.0 then P1 and P2 as calculated are in fact G1 and G2, i.e. the static wheel loading.
Further, in the condition where P1 and P2 = 0, recalculate the angles THT 1 and THT 2 which correspond to the initial front and rear torsion bar wind up angles. Calculate also the static deflection front and rear.
Using subroutine BUMP calculate suspension characteristics for different permutations of bounce and rebound, i.e. different values of F1 and F2 both positive and negative.
For each permutation print
F1, F2, P1, P2, THT1, THT2, THT3, THT4, P/G1, P/G2, RATIO, ITS, C11, C22, C12, C21, PF, BF, PFP, CC1, CC2, PHI1, PHI2, S1, S2 and S3.
END PROGRAM

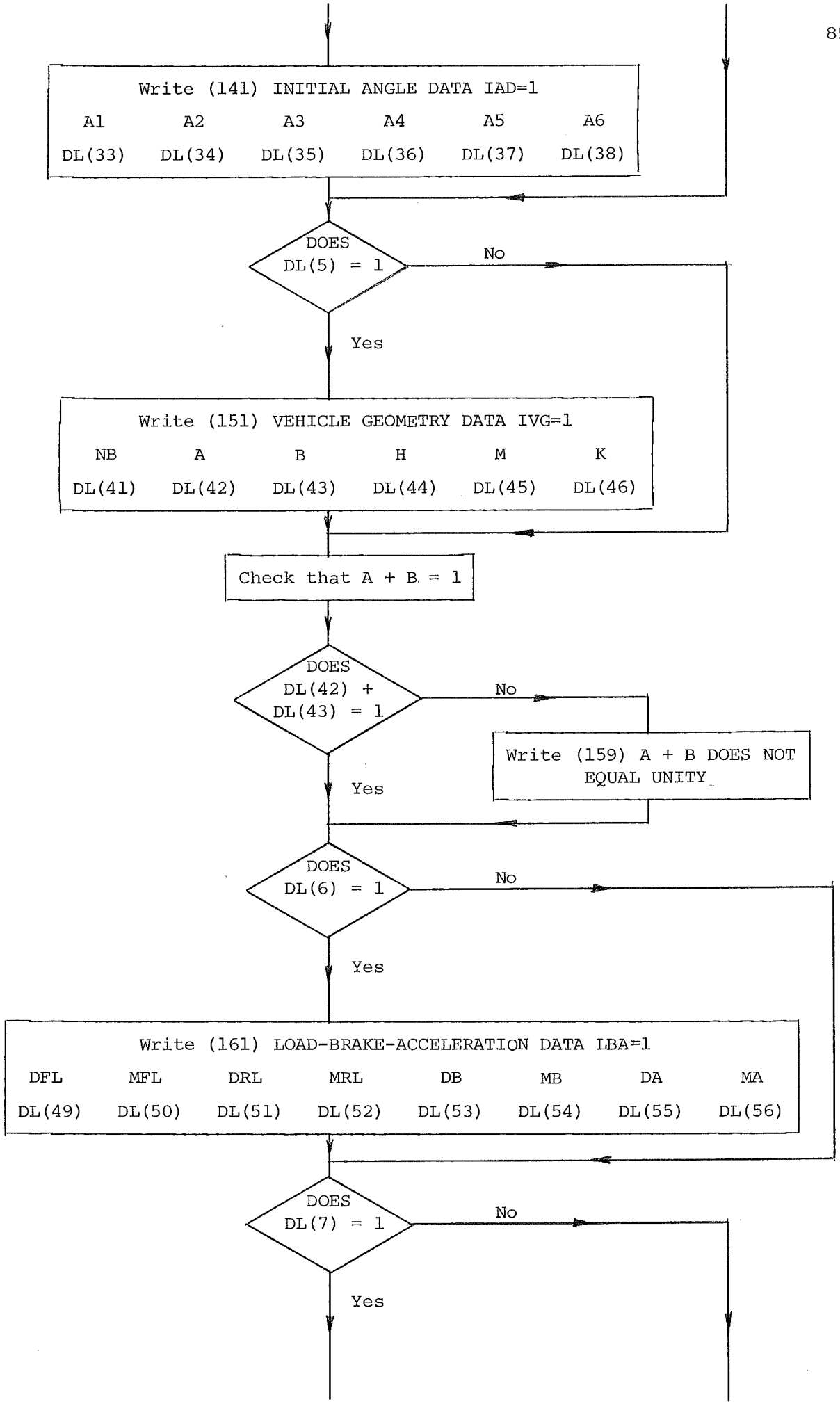
SUBROUTINE SORTDT (DL)

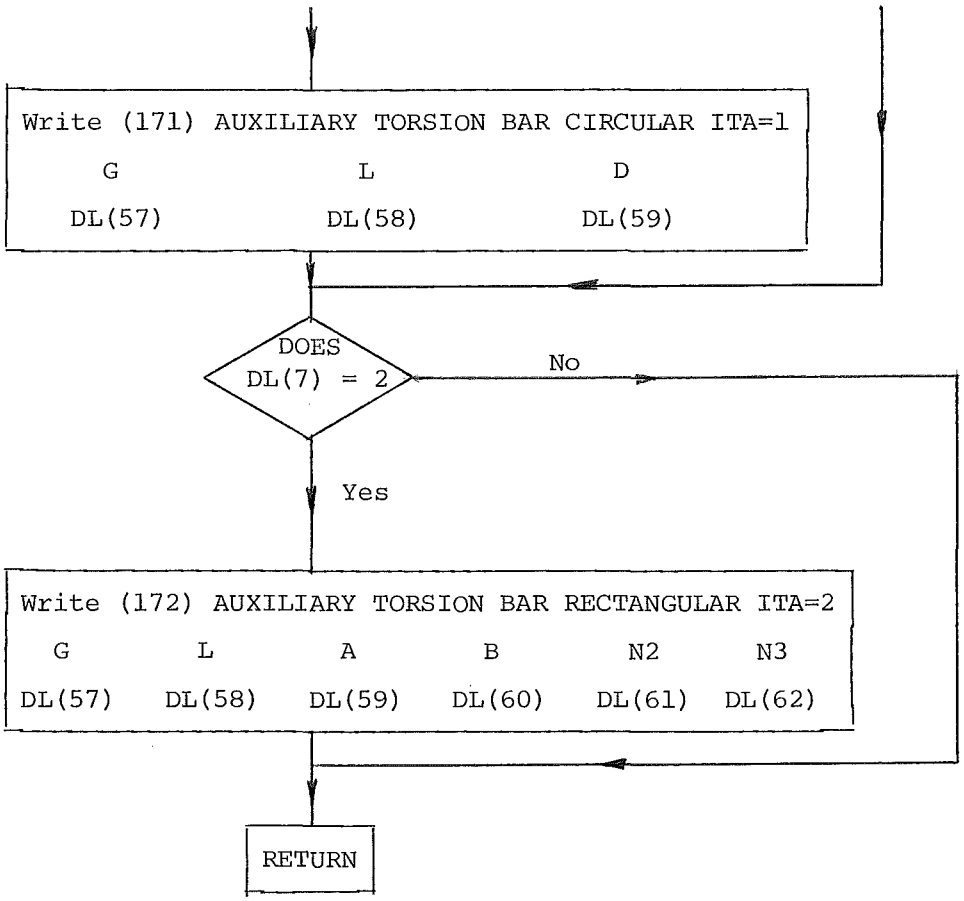
Purpose: To sort and write data.

FLOWCHART




DOES
DL(2) = 1
No
Yes
Write (121) REAR TORSION BAR CIRCULAR ITR=1
G
DL(17)
L
DL(18)
D
DL(19)
DOES
DL(2) = 2
No
Yes
Write (122) REAR TORSION BAR RECTANGULAR ITR=2
G
DL(17)
L
DL(18)
A
DL(19)
B
DL(20)
N2
DL(21)
N3
DL(22)
DOES
DL(3) = 1
No
Yes
Write (131) ARM LENGTH DATE IAL = 1
AL1
DL(25)
AL2
DL(26)
AL3
DL(27)
AL4
DL(28)
AL5
DL(29)
AL6
DL(30)
L3
DL(31)
DOES
DL(4) = 1
No
Yes


Write (141) INITIAL ANGLE DATA IAD=1
A1 A2 A3 A4 A5 A6
DL(33) DL(34) DL(35) DL(36) DL(37) DL(38)
DOES DL(5) = 1
No
Yes
Write (151) VEHICLE GEOMETRY DATA IVG=1
NB A B H M K
DL(41) DL(42) DL(43) DL(44) DL(45) DL(46)
Check that A + B = 1
DOES DL(42) + DL(43) = 1
No
Write (159) A + B DOES NOT EQUAL UNITY
Yes
DOES DL(6) = 1
No
Yes
Write (161) LOAD-BRAKE-ACCELERATION DATA LBA=1
DFL MFL DRL MRL DB MB DA MA
DL(49) DL(50) DL(51) DL(52) DL(53) DL(54) DL(55) DL(56)
DOES DL(7) = 1
No
Yes


Write (171) AUXILIARY TORSION BAR CIRCULAR ITA=1
G
L
D
DL(57)
DL(58)
DL(59)
DOES
DL(7) = 2
No
Yes
Write (172) AUXILIARY TORSION BAR RECTANGULAR ITA=2
G
L
A
B
N2
N3
DL(57)
DL(58)
DL(59)
DL(60)
DL(61)
DL(62)
RETURN

SUBROUTINE TORBAR (DL, ANS)

Purpose. To calculate the torsional springrate in kNm/radian and the stressrate in MPa/radian of the front, rear and auxiliary torsion bars as either circular or rectangular torsion bars.

Input. The relevant data has been read into the "data list" array. By calling the "data list" into the subroutine, no further input is required.

The numeric value of DL(1), DL(2) and DL(7) determine whether the front, rear and auxiliary torsion bars respectively are circular or rectangular.

Output. Having used DL(1), DL(2) and DL(7) as control integers, the appropriate torsional springrate and stressrates are calculated.

	Torsional Rate	Stressrate
Front	ANS(1)	ANS(2)
Rear	ANS(3)	ANS(4)
Auxiliary	ANS(5)	ANS(6)

Discussion

Using the "Manual on Design and Manufacture of Torsion Bar Springs" by Society of Automobile Engineers August 1947.

For a circular torsion bar

Windup angle $\phi = \frac{32\ T\ L}{\pi d^4 G}$ radians

Torsional rate $T' = \frac{T}{\phi} = \frac{\pi d^4 G}{32L}$ Nm/radian

Stress $S = \frac{16T}{\pi d^3} = \frac{\phi d G}{2L}$ Pa

Stress rate $S/\phi = \frac{dG}{2L}$ Pa/radian

For a rectangular torsion bar

Windup angle $\phi = \frac{TL}{\eta_3 a^3 b\ G}$ radians

Torsional rate $T' = \frac{T}{\phi} = \frac{\eta_3 a^3 b\ G}{L}$ Nm/rad.

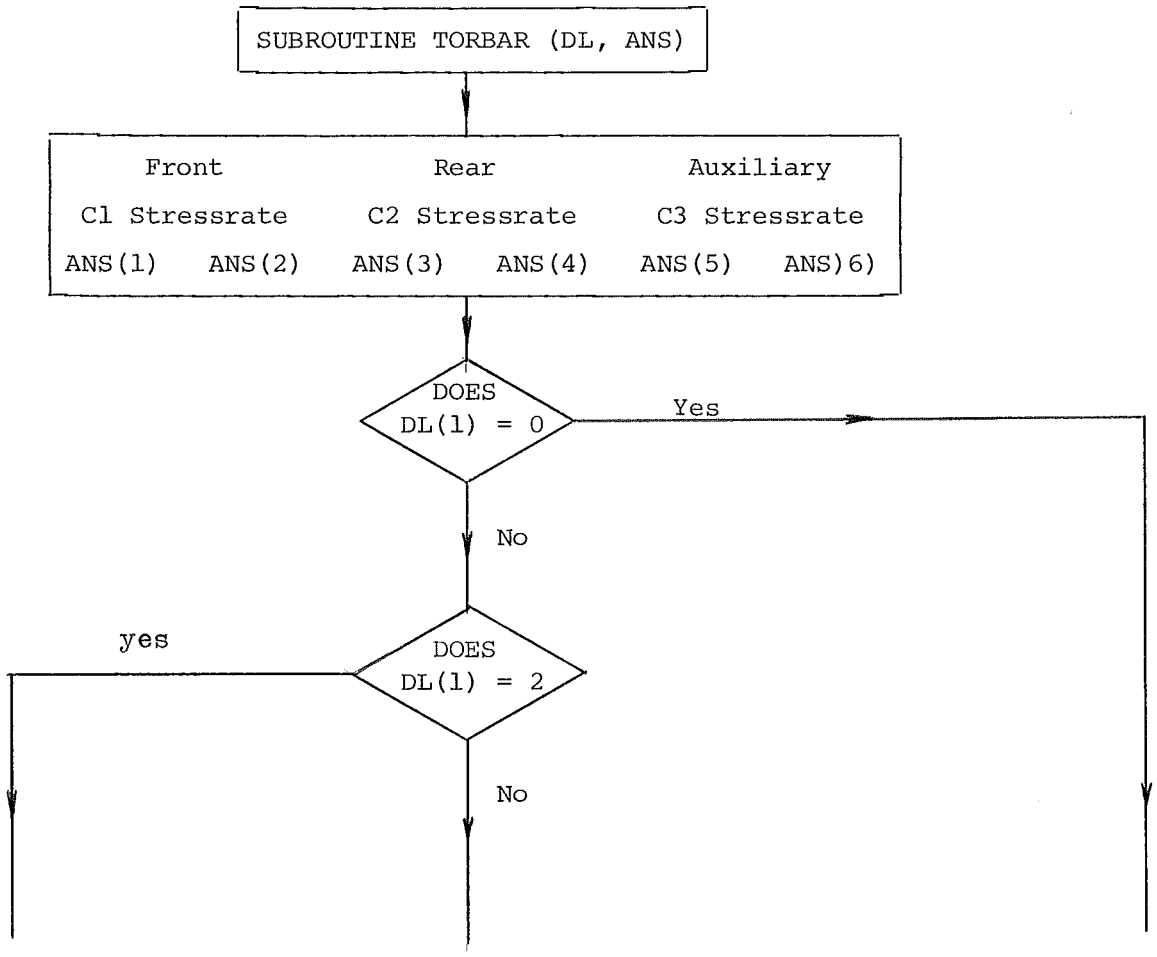
Stress $S = \frac{T}{\eta_2 a^2 b} = \frac{\eta_3\ \phi\ a\ G}{\eta_2\ L}$ Pa

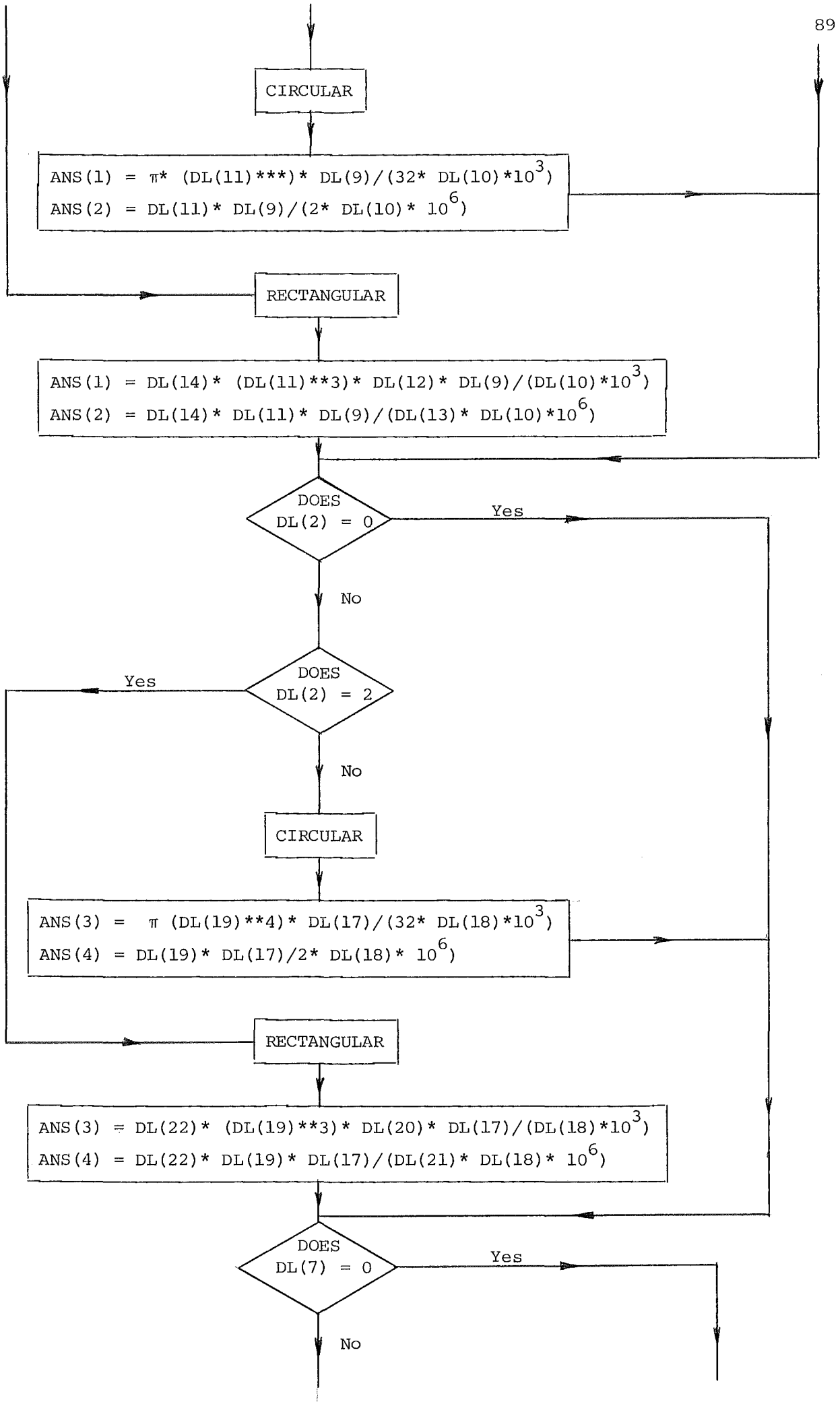
Stress rate $\frac{S}{\phi} = \frac{\eta_3\ a\ G}{\eta_2\ L}$ Pa/radian

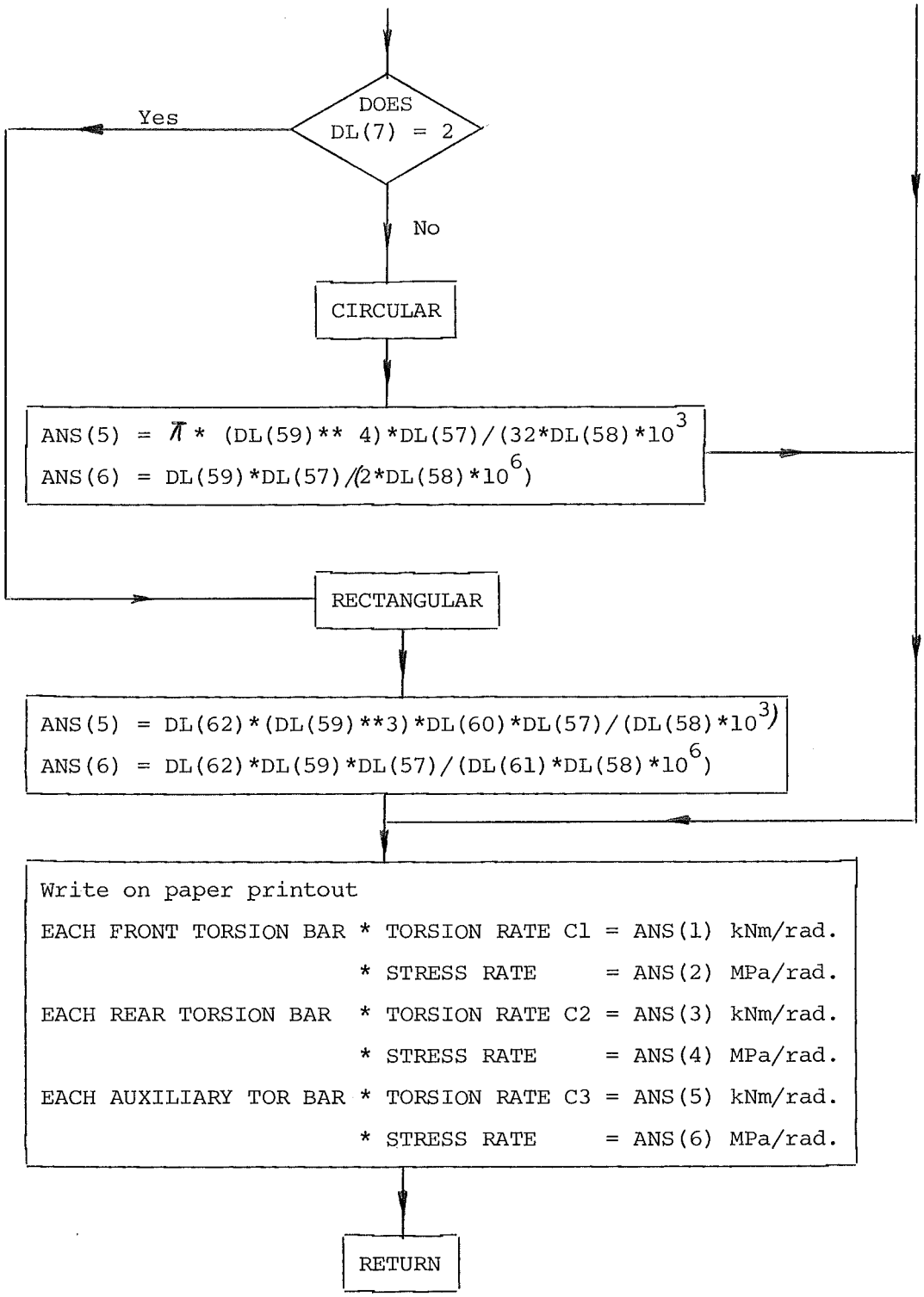
where

T	=	Torque applied at bar	Nm
T'	=	Torsion spring rate of bar	Nm/radian
L	=	Active length of bar	m
d	=	Diameter of round bar	m
a	=	Short side of rectangular bar	m
b	=	Long side of rectangular bar	m
η_2	=	Saint Venant's stress coefficient	
η_3	=	Saint Venant's stiffness coefficient	
ϕ	=	Angle of windup	
S	=	Shear stress	
G	=	Shear modulus.	

FLOWCHART



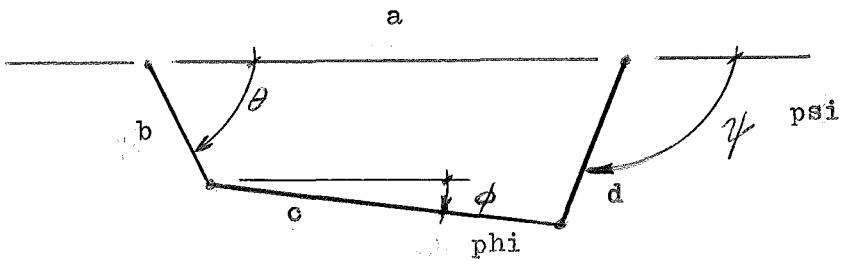

CIRCULAR
ANS(1) = π* (DL(11)***)* DL(9)/(32* DL(10)*10^3)
ANS(2) = DL(11)* DL(9)/(2* DL(10)* 10^6)
RECTANGULAR
ANS(1) = DL(14)* (DL(11)**3)* DL(12)* DL(9)/(DL(10)*10^3)
ANS(2) = DL(14)* DL(11)* DL(9)/(DL(13)* DL(10)*10^6)
DOES DL(2) = 0
Yes
No
DOES DL(2) = 2
Yes
No
CIRCULAR
ANS(3) = π (DL(19)**4)* DL(17)/(32* DL(18)*10^3)
ANS(4) = DL(19)* DL(17)/2* DL(18)* 10^6)
RECTANGULAR
ANS(3) = DL(22)* (DL(19)**3)* DL(20)* DL(17)/(DL(18)*10^3)
ANS(4) = DL(22)* DL(19)* DL(17)/(DL(21)* DL(18)* 10^6)
DOES DL(7) = 0
Yes
No


DOES
DL(7) = 2
Yes
No
CIRCULAR
ANS(5) = π * (DL(59)** 4)*DL(57)/(32*DL(58)*10^3
ANS(6) = DL(59)*DL(57)/(2*DL(58)*10^6)
RECTANGULAR
ANS(5) = DL(62)*(DL(59)**3)*DL(60)*DL(57)/(DL(58)*10^3)
ANS(6) = DL(62)*DL(59)*DL(57)/(DL(61)*DL(58)*10^6)
Write on paper printout
EACH FRONT TORSION BAR * TORSION RATE C1 = ANS(1) kNm/rad.
* STRESS RATE = ANS(2) MPa/rad.
EACH REAR TORSION BAR * TORSION RATE C2 = ANS(3) kNm/rad.
* STRESS RATE = ANS(4) MPa/rad.
EACH AUXILIARY TOR BAR * TORSION RATE C3 = ANS(5) kNm/rad.
* STRESS RATE = ANS(6) MPa/rad.
RETURN

SUBROUTINE FBLANG (K,D,A)

Purpose. To find the two angles of a four bar linkage system, given one angle and the four linkage lengths.

The four lengths are the cable length, the front and rear pitch arms and the pitch arm pivot centres. The input angle is that of the front pitch arm.



Machine Design
June 9 1977 p 94.

$$\phi = \pm \cos^{-1}\left\{\frac{(a^2 + b^2 - 2ab\cos\theta) + c^2 - d^2}{2c\sqrt{a^2+b^2 - 2ab\cos\theta}}\right\} - \tan^{-1}\left(\frac{b\sin\theta}{a-b\cos\theta}\right)$$

$$\psi = 180 \pm \cos^{-1}\left\{\frac{(a^2+b^2-2ab\cos\theta)+d^2-c^2}{2d\sqrt{a^2+b^2 - 2ab\cos\theta}}\right\} - \tan^{-1}\left(\frac{b\sin\theta}{a-b\cos\theta}\right)$$

FBLANG means Four Bar Linkage Angles.

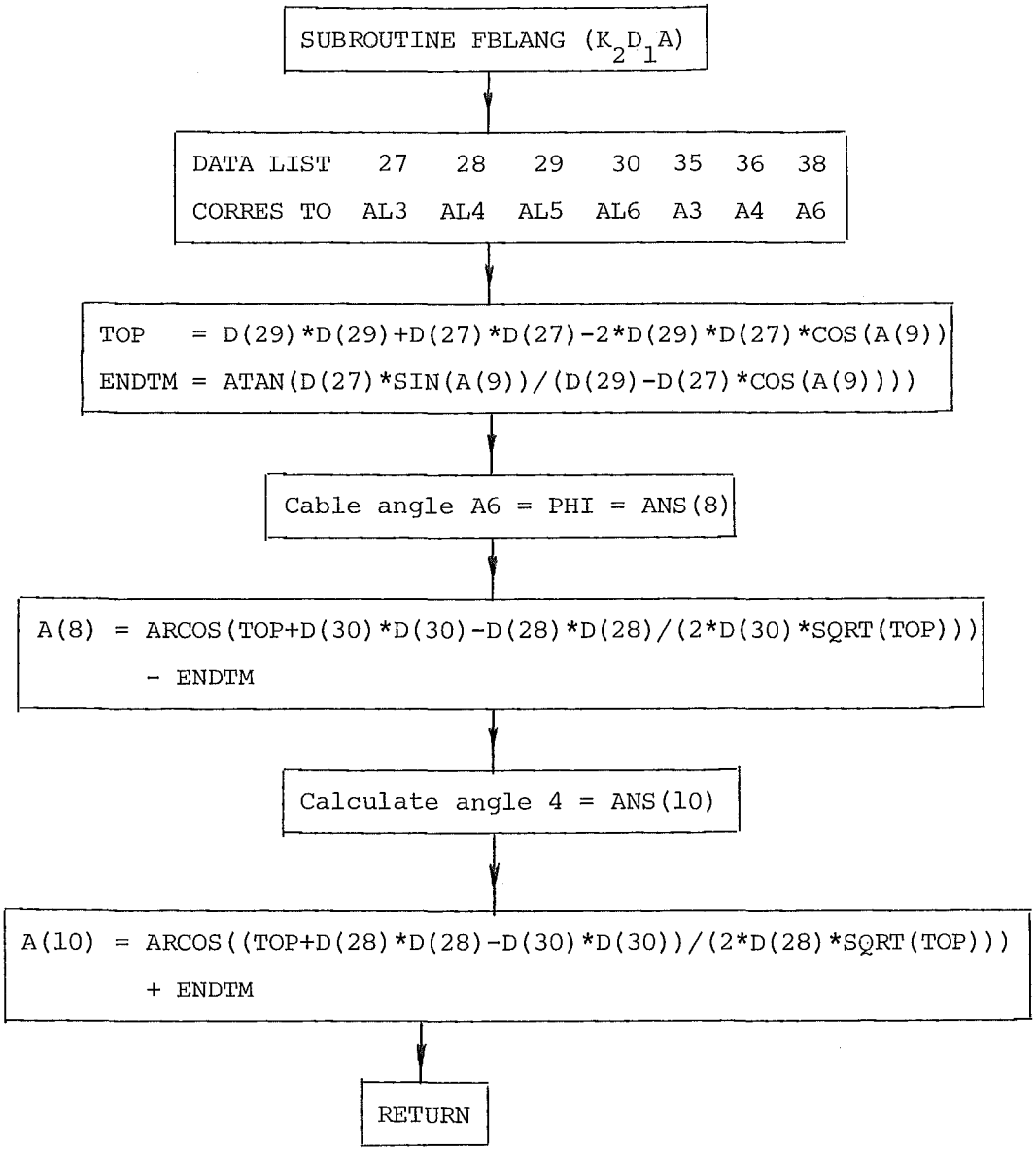
Input. K - control integer
DL - data list array
ANS - answer array.

subroutine uses ANS(9) DL(27) (28) (29) (30)
which are A3 AL3 AL4 AL5 AL6

Note:- Within this subroutine the arrays data list DL and answers ANS are referred to as arrays D and A respectively.

Output. ANS(8) the cable angle A6.
ANS(10) the angle of the rear pitch arm A4.

FLOWCHART

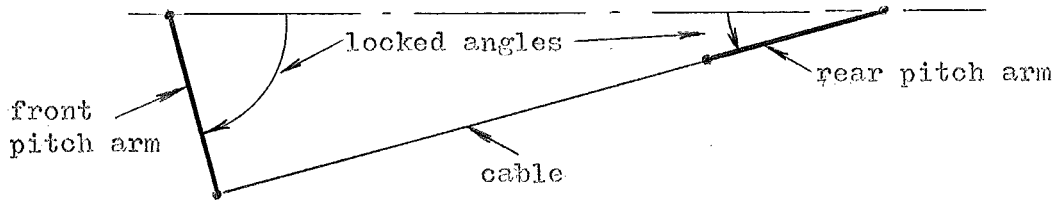

SUBROUTINE FBLANG (K_2 D_1 A)
DATA LIST 27 28 29 30 35 36 38
CORRES TO AL3 AL4 AL5 AL6 A3 A4 A6
TOP = D(29)*D(29)+D(27)*D(27)-2*D(29)*D(27)*COS(A(9))
ENDTM = ATAN(D(27)*SIN(A(9))/(D(29)-D(27)*COS(A(9))))
Cable angle A6 = PHI = ANS(8)
A(8) = ARCOS(TOP+D(30)*D(30)-D(28)*D(28)/(2*D(30)*SQRT(TOP)))
- ENDTM
Calculate angle 4 = ANS(10)
A(10) = ARCOS((TOP+D(28)*D(28)-D(30)*D(30))/(2*D(28)*SQRT(TOP)))
+ ENDTM
RETURN

SUBROUTINE ARMLTH (DL,ANS)

Purpose. To calculate the actual length of the front and rear pitch arms and the length of the connecting cable to enable further calculations in the bounce/rebound configuration to be made.

The initial cable angle is also calculated as is the locked pitch angles. These locked angles are calculated maximum angles that the pitch angles can attain during a bump or rebound condition and are used as a check further in the program.

The pitch arms are deemed to be locked when the front or rear arm and the cable are in a straight line.



Locked Pitch Angles - front bump.

Input. DL - data list array

ANS - answer array.

subroutine uses DL 25 26 27 28 29 30 31 33 34 35 36 38

which correspond to AL1 AL2 AL3 AL4 AL5 AL6 L3 A1 A2 A3 A4 A6

42 43

A B

Output. ANS (7) = RL34 = initial ratio ℓ_3/ℓ_4.

ANS(11) = Front locked pitch angle front bump.

ANS(12) = Rear locked pitch angle front bump.

ANS(13) = Front locked pitch angle, rear bump.

ANS(14) = Rear locked pitch angle, rear bump.

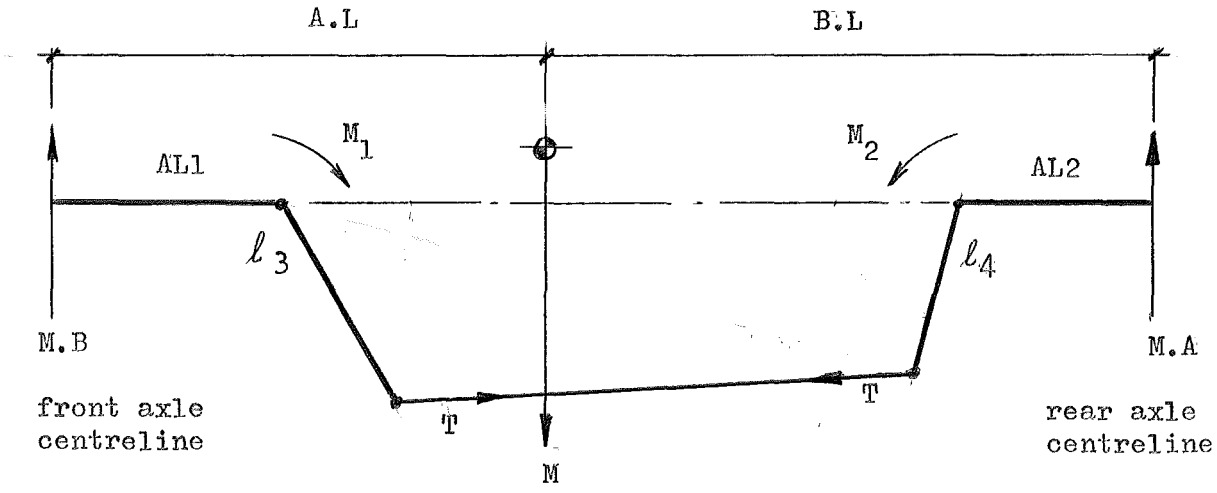
ANS(23) = Effective length ℓ_3.

ANS(24) = Effective length ℓ_4.

Discussion

Formula used in subroutine ARMLTH are, in sequent order, are as follows.

Ratio of lengths ℓ_3 and ℓ_4, from the figure below,



where

M = vehicle mass through C of G.

M_1 = moment at torsion bar 1.

M_2 = moment at torsion bar 2.

ℓ_3, ℓ_4 = effective length.

T = cable tension

then

M_1 = M.B. ALI. Cos(A1) = T.ℓ_3

M_2 = M.A. AL2. Cos(A2) = T.ℓ_4

and

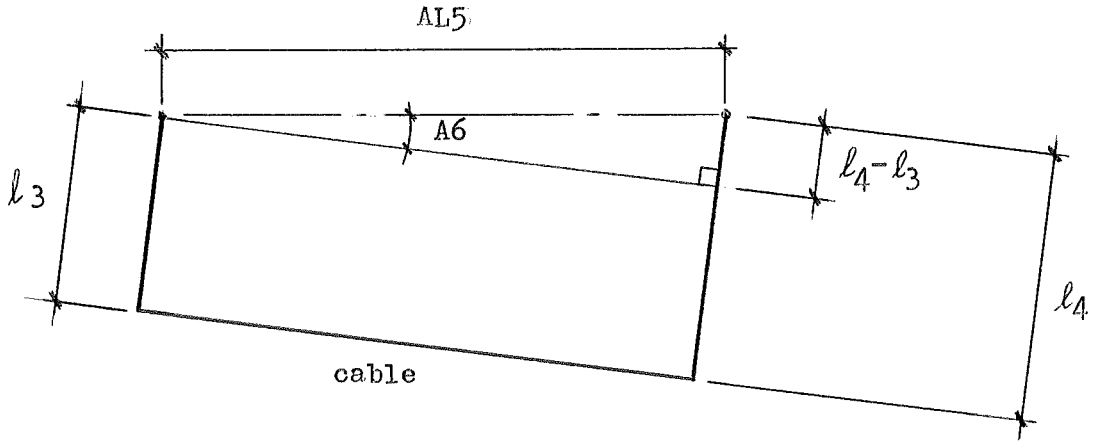
$$\ell_3/\ell_4 = RL34 = ANS(7) = \frac{B_1\ ALl_1\ \cos\ (A1)}{A.\ AL2.\ \cos\ (A2)}$$

so that

ANS(23) = ℓ_3 = DL(31) a data input

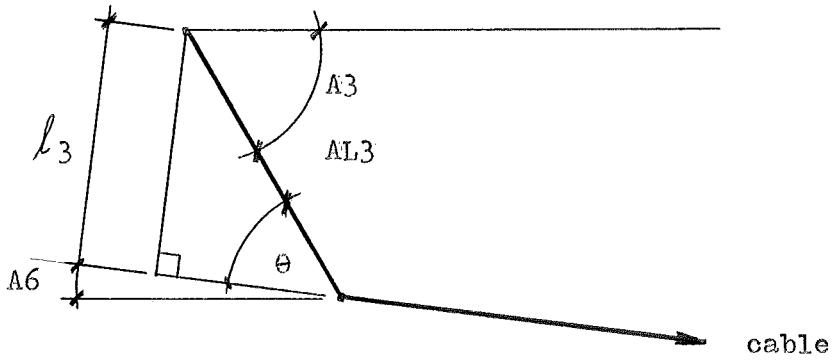
$$ANS(24) = \ell_4 = \frac{ANS(23)}{ANS(7)} = \frac{\ell_3}{RL34}$$

Cable angle A6 = DL(38) = $\sin^{-1}\left(\frac{\ell_4-\ell_3}{AL5}\right)$



Actual front pitch arm length

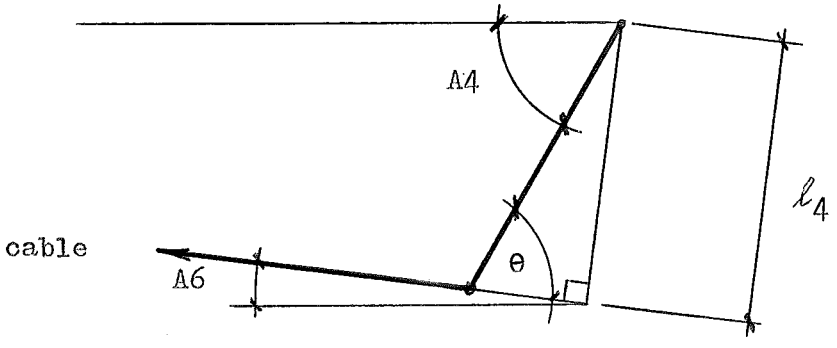
AL3 = DL(27) = $\ell_3/\sin(A3-A6)$



θ=A3-A6

Actual rear pitch arm length

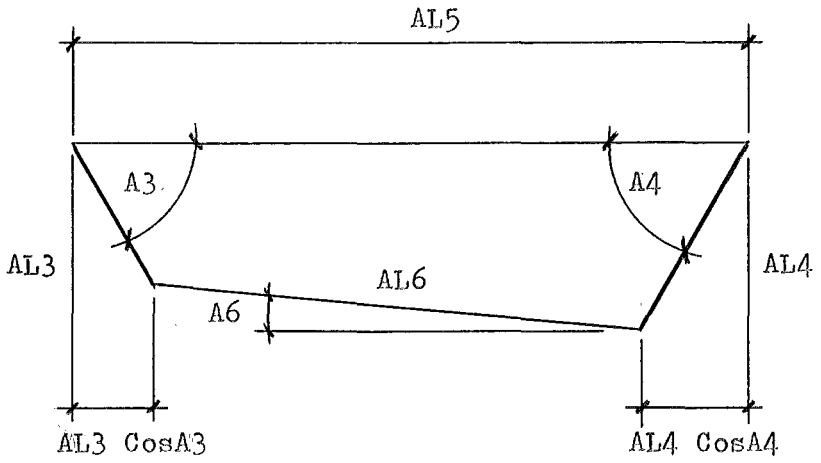
AL4 = DL(28) = $\ell_4/\sin(A4+A6)$



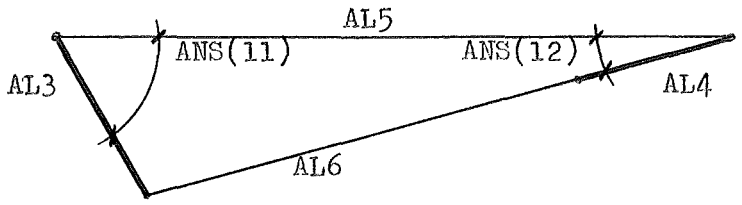
θ=A4+A6

Actual cable length

$$DL(30) = AL6 = \frac{(AL5-AL3\ \cos(A3)-AL4\ \cos(A4))}{\cos(A6)}$$



Calculate S = ½ the perimeter of the triangle shown below.



Front Bump Locked Pitch Angles.

Front locked angle = ANS(11)

$$ANS(11) = 2\ ARSIN\left(\sqrt{\frac{(S-AL3)(S-AL5)}{AL3.AL5}}\right)$$

Corresponding locked rear angle.

$$ANS(12) = 2\ ARCOS\left(\sqrt{\frac{S(S-AL3)}{AL5(AL6+AL4)}}\right)$$

Similarly for rear bump locked pitch arms.

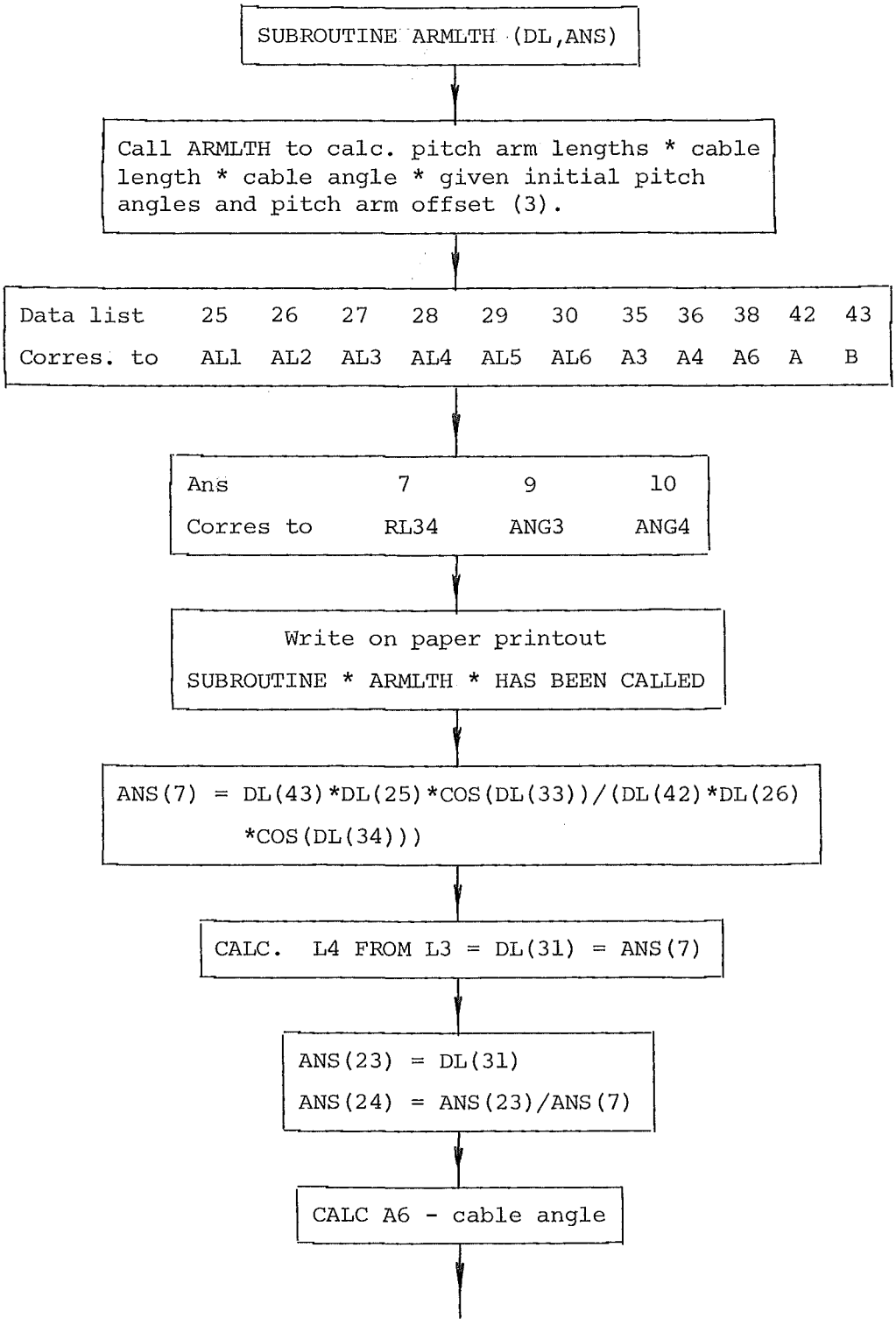
Front locked angle

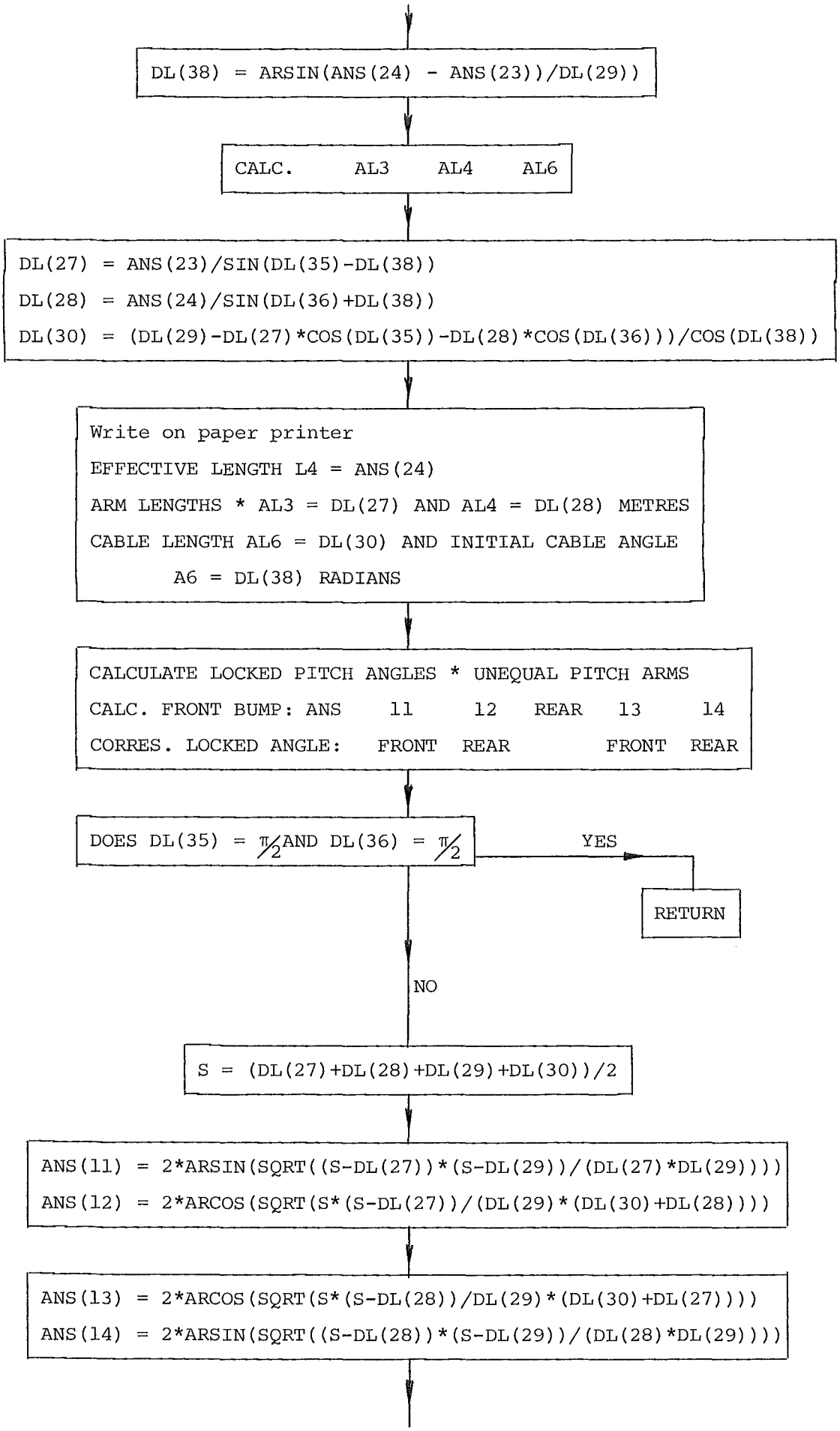
$$ANS(13) = 2.ARCOS\left(\sqrt{\frac{S(S-AL4)}{AL5(AL6+AL3)}}\right)$$

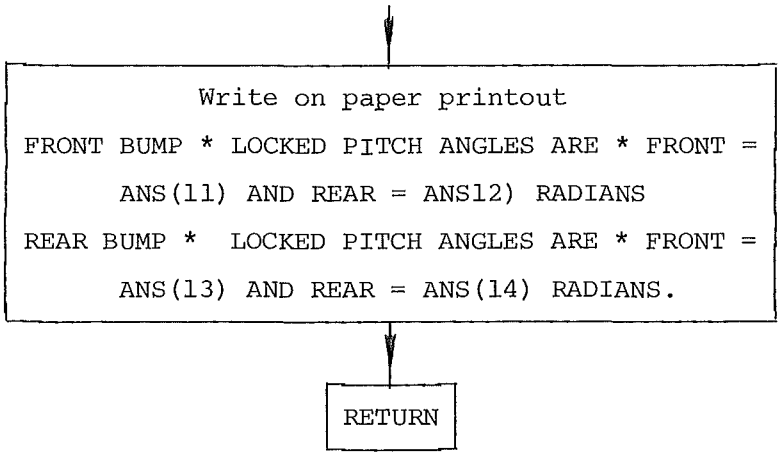
Corresponding rear locked angle

$$ANS(14) = 2.ARSIN\left(\sqrt{\frac{(S-AL4)(S-AL5)}{AL4.AL5}}\right)$$

FLOWCHART




DL(38) = ARSIN(ANS(24) - ANS(23))/DL(29))
CALC. AL3 AL4 AL6
DL(27) = ANS(23)/SIN(DL(35)-DL(38))
DL(28) = ANS(24)/SIN(DL(36)+DL(38))
DL(30) = (DL(29)-DL(27)*COS(DL(35))-DL(28)*COS(DL(36)))/COS(DL(38))
Write on paper printer
EFFECTIVE LENGTH L4 = ANS(24)
ARM LENGTHS * AL3 = DL(27) AND AL4 = DL(28) METRES
CABLE LENGTH AL6 = DL(30) AND INITIAL CABLE ANGLE
A6 = DL(38) RADIANS
CALCULATE LOCKED PITCH ANGLES * UNEQUAL PITCH ARMS
CALC. FRONT BUMP: ANS 11 12 REAR 13 14
CORRES. LOCKED ANGLE: FRONT REAR FRONT REAR
DOES DL(35) = π/2 AND DL(36) = π/2
YES
RETURN
NO
S = (DL(27)+DL(28)+DL(29)+DL(30))/2
ANS(11) = 2*ARSIN(SQRT((S-DL(27))*(S-DL(29))/(DL(27)*DL(29))))
ANS(12) = 2*ARCOS(SQRT(S*(S-DL(27))/(DL(29)*(DL(30)+DL(28)))))
ANS(13) = 2*ARCOS(SQRT(S*(S-DL(28))/DL(29)*(DL(30)+DL(27))))
ANS(14) = 2*ARSIN(SQRT((S-DL(28))*(S-DL(29))/(DL(28)*DL(29))))


Write on paper printout
FRONT BUMP * LOCKED PITCH ANGLES ARE * FRONT =
ANS(11) AND REAR = ANS12) RADIANS
REAR BUMP * LOCKED PITCH ANGLES ARE * FRONT =
ANS(13) AND REAR = ANS(14) RADIANS.
RETURN

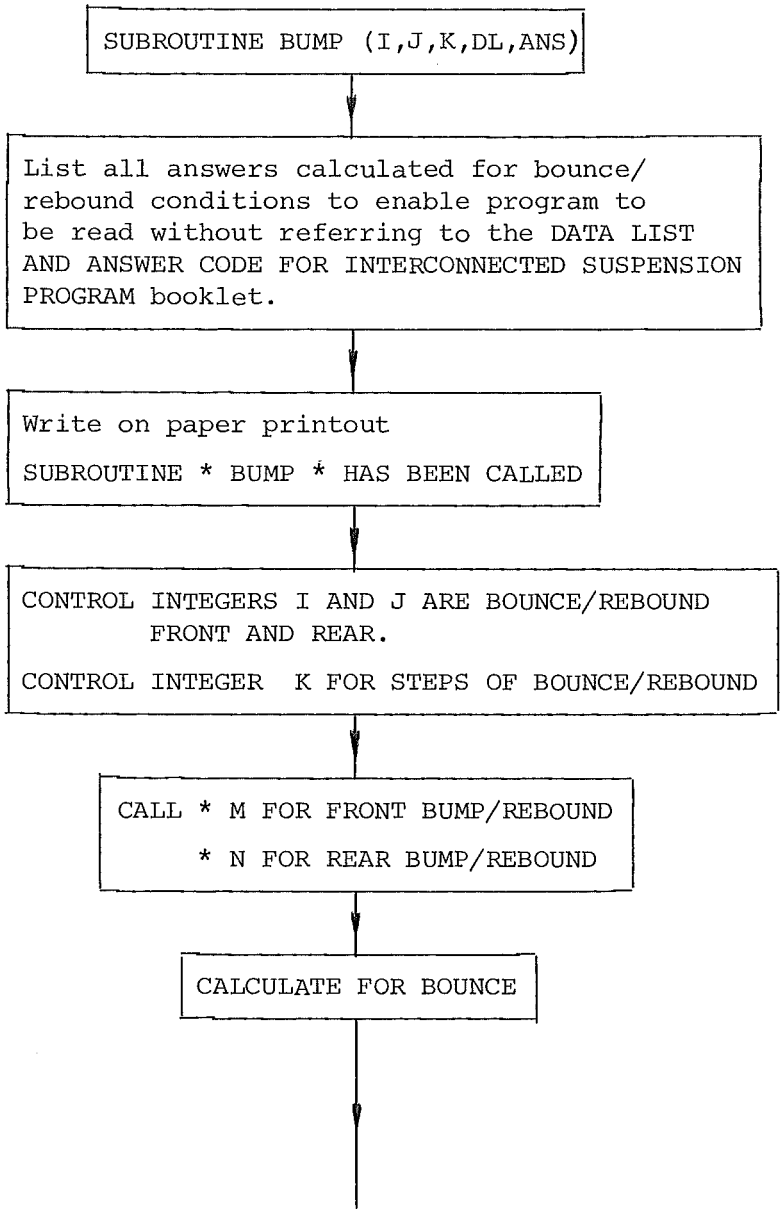
SUBROUTINE BUMP (I,J,K,DL,ANS)

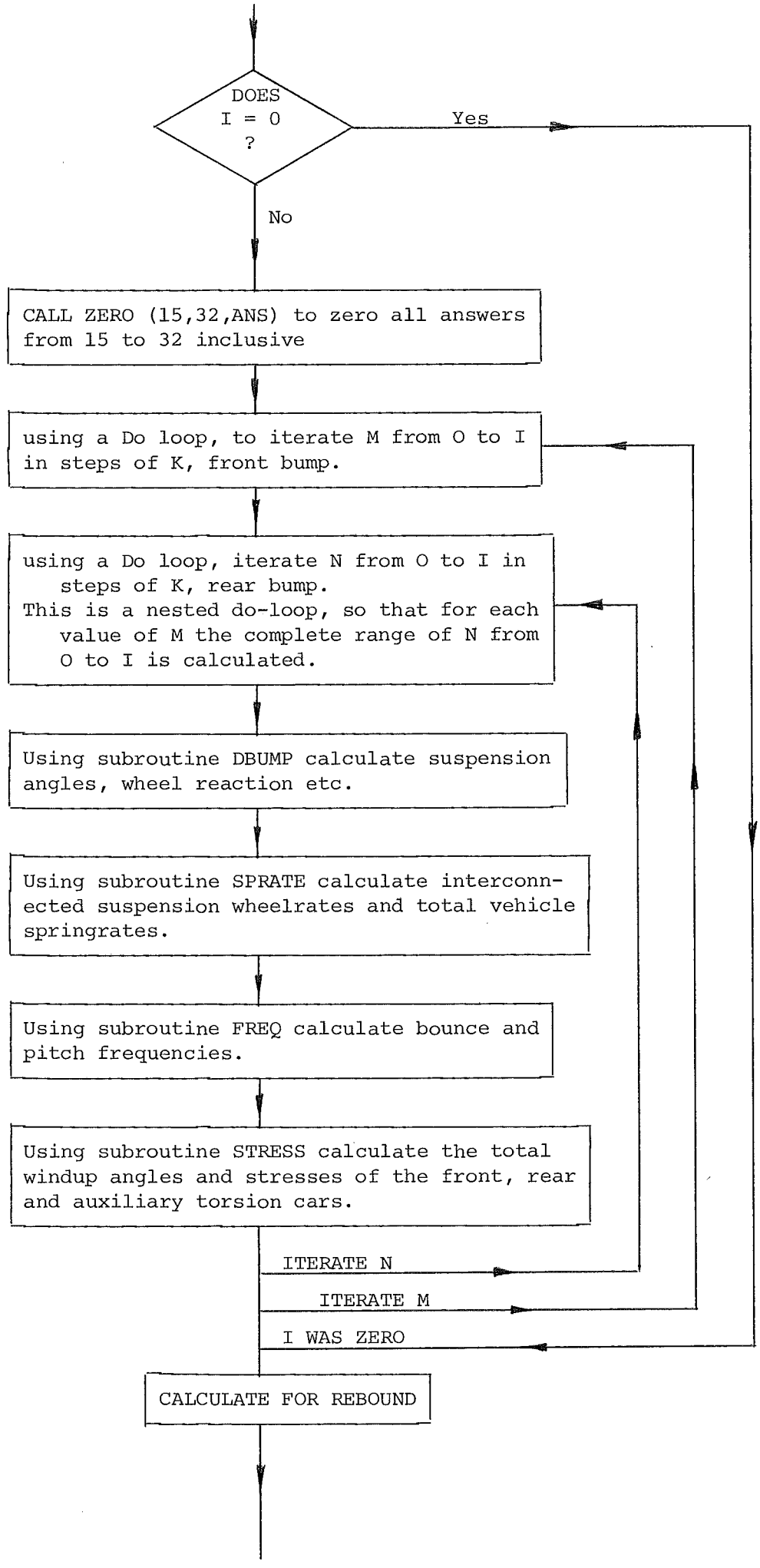
Purpose. Once subroutine BUMP has been called in the main program, this subroutine calls the necessary subroutines to calculate, for all the required bounce/rebound permutations, the wheel reactions, springrates, frequencies and torsion bar stresses.

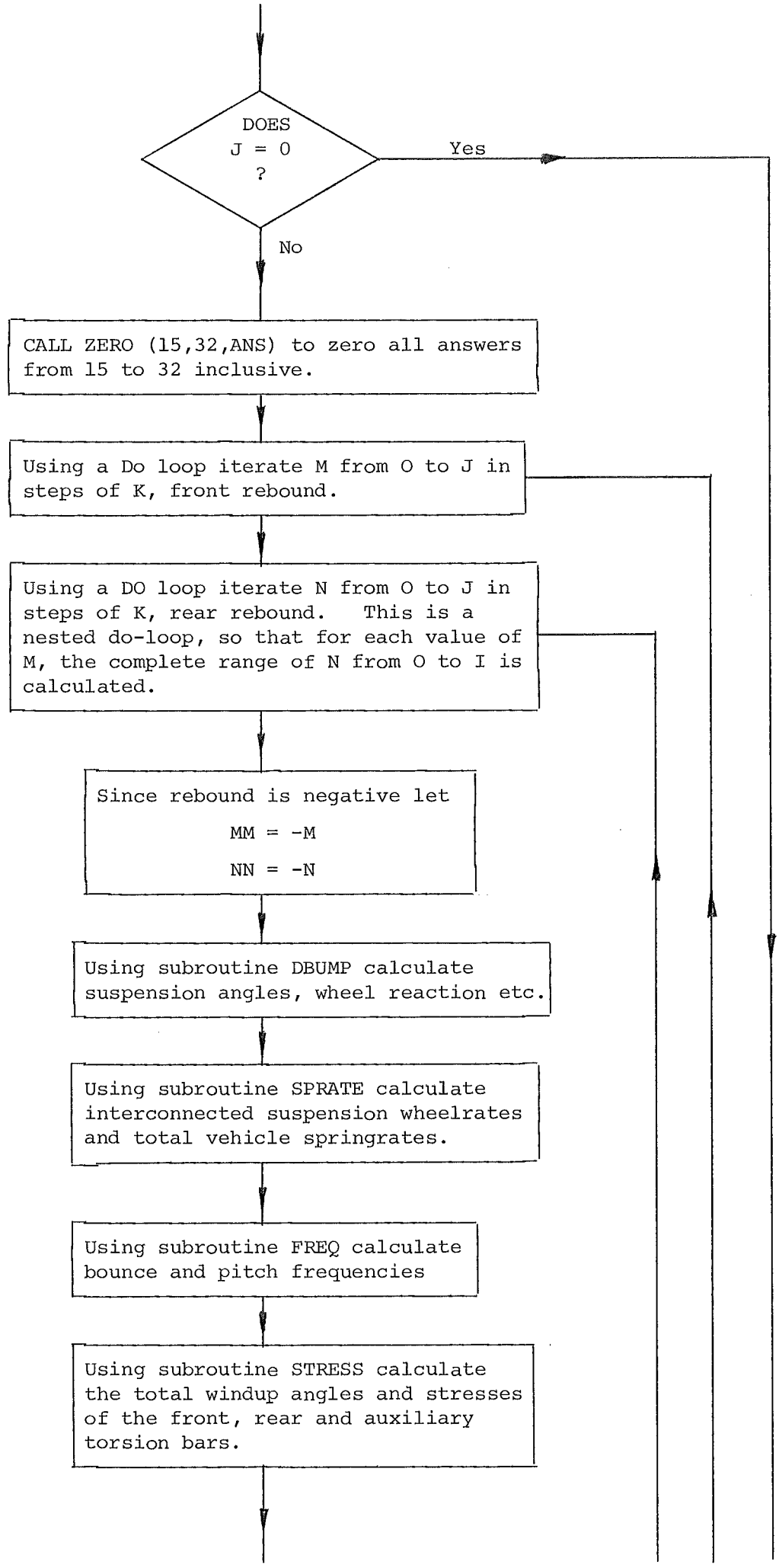
Input.
- I = Maximum bounce mm.
- J = Maximum rebound mm.
- K = Steps for I/J iteration mm.
- DL = Data List Array.
- ANS = Answers Array.

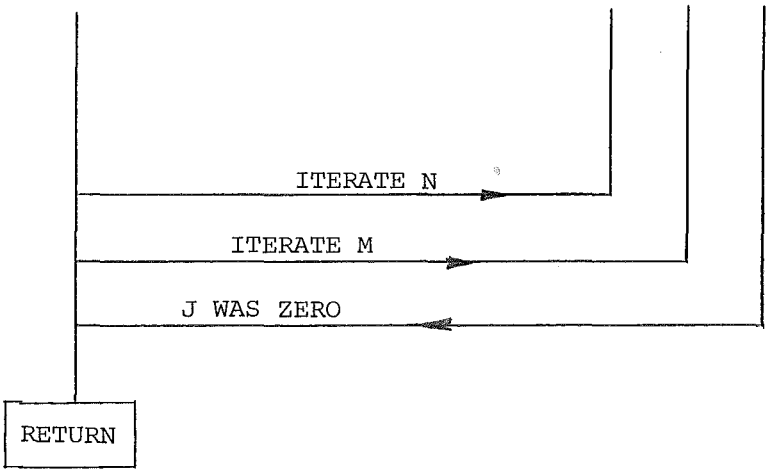
Output. All information that is required for each bounce/rebound condition, calculated in subroutines DBUMP, SPRATE, FREQ and STRESS.

FLOWCHART




DOES I = 0 ?
Yes
No
CALL ZERO (15,32,ANS) to zero all answers from 15 to 32 inclusive
using a Do loop, to iterate M from O to I in steps of K, front bump.
using a Do loop, iterate N from O to I in steps of K, rear bump.
This is a nested do-loop, so that for each value of M the complete range of N from O to I is calculated.
Using subroutine DBUMP calculate suspension angles, wheel reaction etc.
Using subroutine SPRATE calculate interconnected suspension wheelrates and total vehicle springrates.
Using subroutine FREQ calculate bounce and pitch frequencies.
Using subroutine STRESS calculate the total windup angles and stresses of the front, rear and auxiliary torsion cars.
ITERATE N
ITERATE M
I WAS ZERO
CALCULATE FOR REBOUND


DOES
J = 0
?
Yes
No
CALL ZERO (15,32,ANS) to zero all answers from 15 to 32 inclusive.
Using a Do loop iterate M from O to J in steps of K, front rebound.
Using a DO loop iterate N from O to J in steps of K, rear rebound. This is a nested do-loop, so that for each value of M, the complete range of N from O to I is calculated.
Since rebound is negative let
MM = -M
NN = -N
Using subroutine DBUMP calculate suspension angles, wheel reaction etc.
Using subroutine SPRATE calculate interconnected suspension wheelrates and total vehicle springrates.
Using subroutine FREQ calculate bounce and pitch frequencies
Using subroutine STRESS calculate the total windup angles and stresses of the front, rear and auxiliary torsion bars.


ITERATE N
ITERATE M
J WAS ZERO
RETURN

SUBROUTINE DBUMP (M, N, DL, ANS)

Purpose. To calculate all the mechanical detail for any front and rear (M and N) bump and rebound (+ve and -ve, M and N) condition to enable the frequencies and torsion bar stresses to be subsequently calculated.

Input. M = front bump or rebound bump +ve, rebound -ve
dimensions in mm.
N = rear bump or rebound bump +ve, rebound -ve
dimensions in mm.
DL = Data list array.
ANS = Answer array.

Subroutine uses

DL(25) = AL1 = Arm length 1, front wheel to torsion bar.
DL(26) = AL2 = Arm length 2, rear wheel to torsion bar.
DL(27) = AL3 = Arm length 3, front pitch arm length.
DL(28) = AL4 = Arm length 4, rear pitch arm length.
DL(33) = A1 = Initial angle of front arm 1.
DL(34) = A2 = Initial angle of rear arm 2.
DL(35) = A3 = Initial angle of pitch arm 3.
DL(36) = A4 = Initial angle of pitch arm 4.
DL(42) = A = Distance front axle to C of G as ratio of wheelbase.
DL(43) = B = Distance rear axle to C of G as ratio of wheelbase.
DL(45) = M = Vehicle mass kg.
ANS(1) = C_1 = Front torsion bar springrate.
ANS(3) = C_2 = Rear torsion bar springrate.
ANS(5) = C_3 = Auxiliary torsion bar springrate.
ANS(8) = A_6 = Cable angle.
ANS(9) = A_3 = Front pitch arm angle.
ANS(10) = A_4 = Rear pitch arm angle.
ANS(51) = G_1 = Static front wheel loading.
ANS(52) = G_2 = Static rear wheel loading.

Output.

Subroutine calculates the following,

ANS(15) = A_1 = Instantaneous angle of arm 1.
ANS(16) = A_2 = Instantaneous angle of arm 2.
ANS(21) = ℓ_1 = Effective length of arm 1.
ANS(22) = ℓ_2 = Effective length of arm 2.

ANS(23) = ℓ_3 = Effective length of arm 3.

ANS(24) = ℓ_4 = Effective length of arm 4.

ANS(25) = RL34 = Ratio lengths ℓ_3/ℓ_4.

ANS(26) = RA34 = Ratio angles θ_3/θ_4.

Subroutine calculates and prints the following,

ANS(17) = θ_1 = Angle deviation from static for arm 1.

ANS(18) = θ_2 = Angle deviation from static for arm 2.

ANS(19) = θ_3 = Angle deviation from static for arm 3.

ANS(20) = θ_4 = Angle deviation from static for arm 4.

ANS(27) = RATIO = RL34/RA34 = ℓ_3/ℓ_4. θ_4/θ_3.

ANS(28) = T = Cable tension.

ANS(29) = F_1 = Front wheel displacement.

ANS(30) = F_2 = Rear wheel displacement.

ANS(31) = P_1 = Front wheel load reaction.

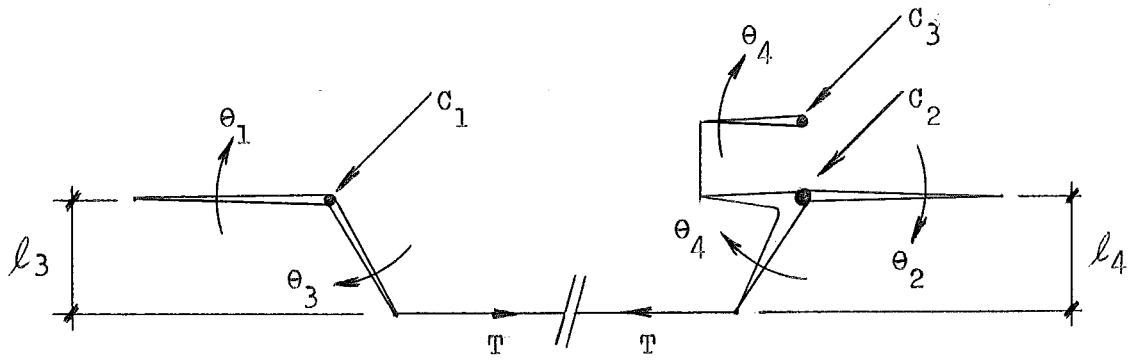
ANS(32) = P_2 = Rear wheel load reaction.

Description.

This subroutine does the background work for the Interconnected Suspension Program, it calculates the position of all the arms, then, from these, the instantaneous front and rear wheel loadings as well as the cable tension (used for cable size selection and chassis loading) for each and every permutation of bounce and rebound, front and rear within the limits specified in the main program.

The subroutine from the wheel displacements inputted, first calculates the angular displacement of the front and rear suspension arms, which is the rotation of the outboard and of the torsion bars, namely θ_1 and θ_2.

The inboard end of the torsion bars, connected to the pitch arms which are in turn connected to each other via the cable, must come to an equilibrium position. This equilibrium is dependent on the lengths and angles of the pitch arms and the cable and on the rotational springrate of the front, rear and auxiliary torsion bars.



θ_1 is the positive for front bounce.

θ_2 is negative for rear bounce.

C_1, C_2, C_3 are torsional springrate for front, rear and auxiliary torsion bars respectively.

ℓ_3, ℓ_4 are effective arm lengths.

Front torsion bar, cable tension T is

$$T = \frac{C_1(\theta_1-\theta_3)}{\ell_3}$$

Rear torsion bar, cable tension T is

$$T = \frac{-\left[C_2(\theta_2-\theta_4) + C_3(0-\theta_4)\right]}{\ell_4}$$

$$T = \frac{-C_2\theta_2 + (C_2+C_3)\theta_4}{\ell_4}$$

Equating the two expressions of T, to find θ_3

$$\theta_3 = \theta_1 + \frac{\ell_3/\ell_4(C_2\theta_2 - (C_2+C_3)\theta_4)}{C_1}$$

However, ℓ_3, ℓ_4 and θ_4 are all dependent on θ_3, either directly or indirectly, so that θ_3 (and hence θ_4 etc.) must be found by iteration.

Using the notation θ for θ_3 so that θ_n is the nth iteration toward finding θ.

θ_n strictly means $(\theta_3)_n$

Given θ_n, we can calculate from equilibrium considerations, as above, what θ_n should be, that is, have θ_n can calculate $f(\theta_n)$ such that they should be equal.

Define $F(\theta_n) = f(\theta_n) - \theta_n = 0$ and the Newton-Raphson Method of Successive Approximations can be used,

$$\theta_{n+1} = \theta_n - \frac{F(\theta_n)}{F'(\theta_n)}$$

If $F(\theta_n) = f(\theta_n) - \theta_n$ then $F'(\theta_n) = f'(\theta_n) - 1$, and $f'(\theta_n)$ can be approximated as,

$$f'(\theta_n) \simeq \frac{f(\theta_n) - f(\theta_{n-1})}{\theta_n - \theta_{n-1}}$$

and making an assumption that

$$f(\theta_{n-1}) = \theta_n$$

though not strictly correct, it does not slow convergence appreciably, so that

$$f'(\theta_n) = \frac{f(\theta_n) - \theta_n}{\theta_n - \theta_{n-1}}$$

then $\theta_{n+1} = \theta_n + \dfrac{f(\theta_n) - \theta_n}{1 - f'(\theta_n)}$

or

$$\text{THTNP1} = \theta + \frac{\text{FTHT} - \theta}{1 - S}$$

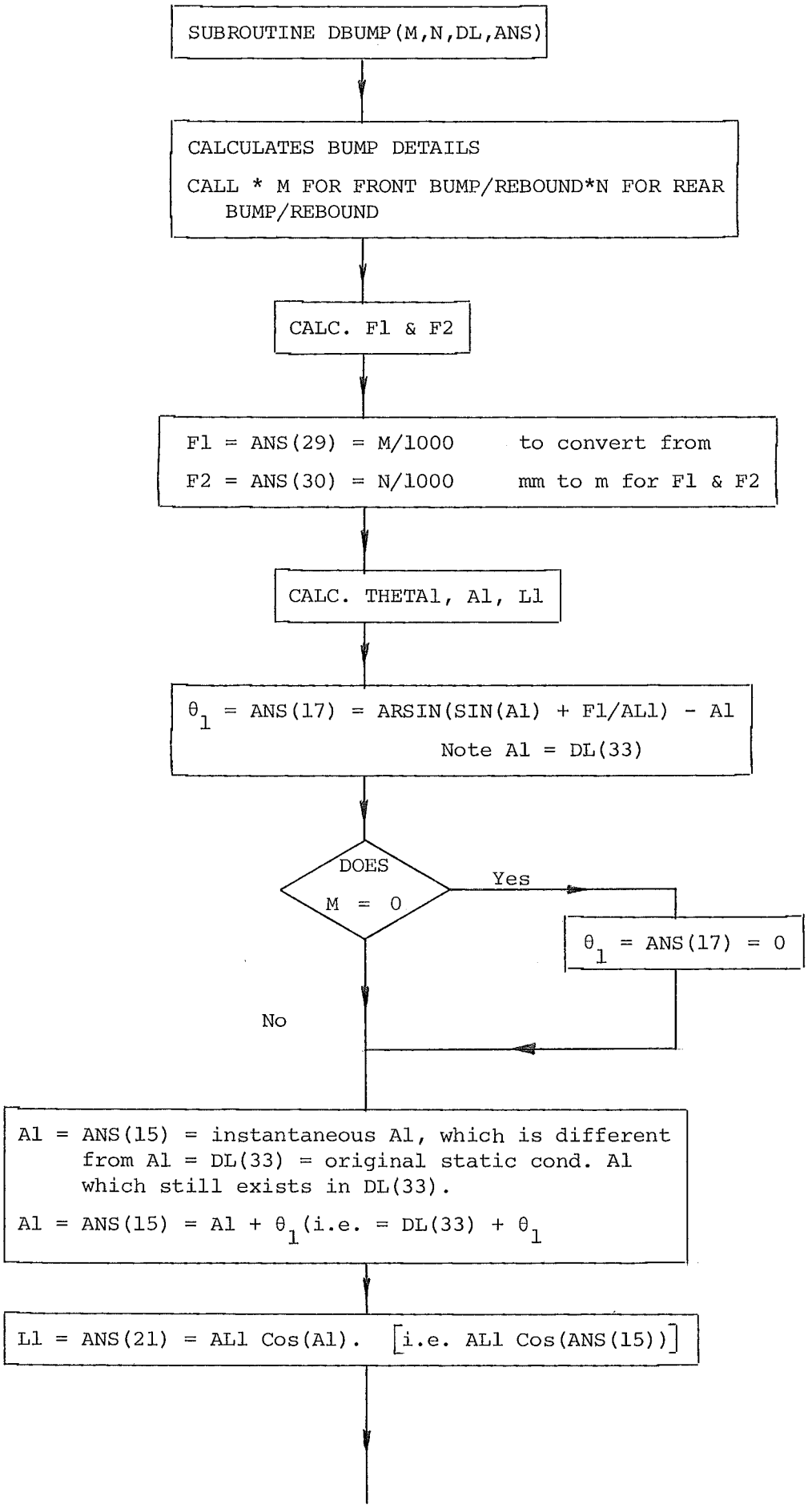
where $$S = \frac{\text{FTHT} - \theta}{\theta - \text{THTNMI}}$$

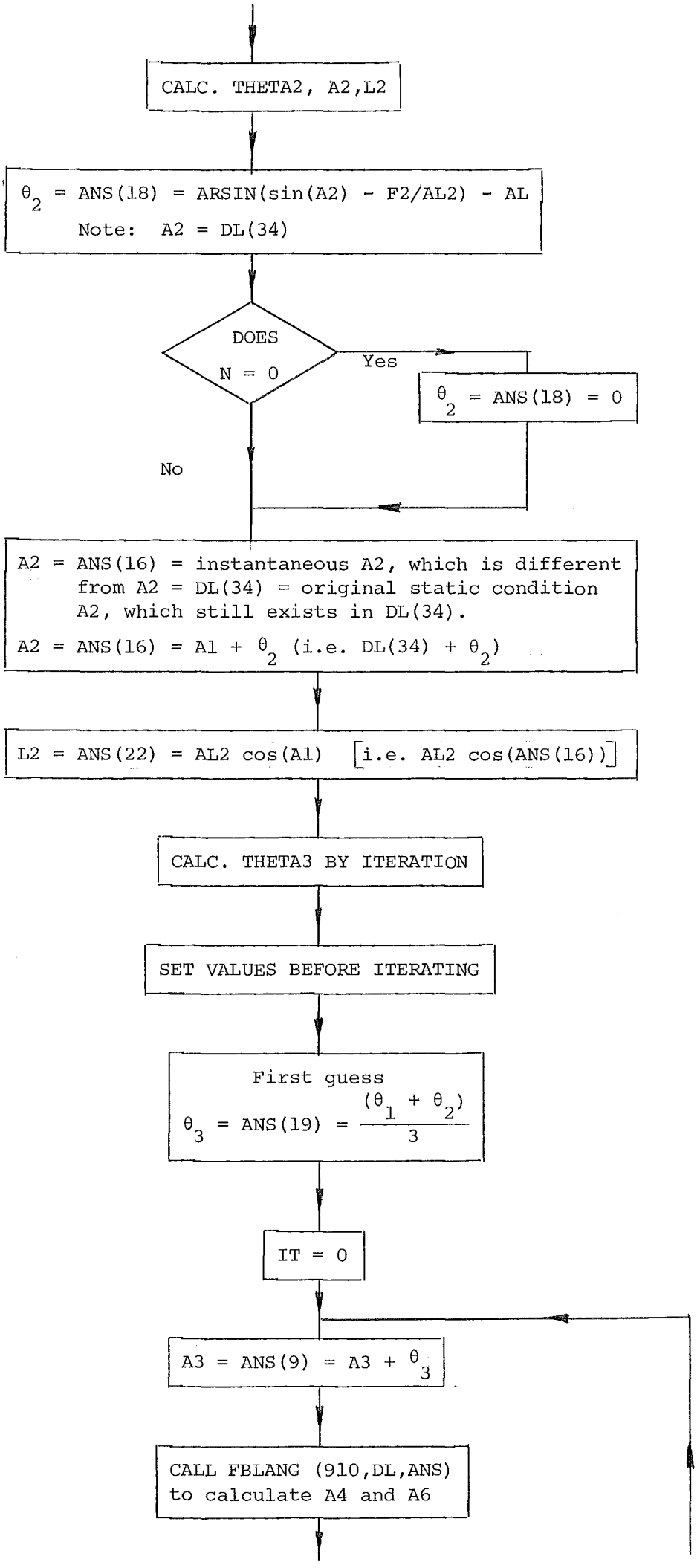
Iterations are stopped when $f(\theta_n) = \theta_n$, i.e. when

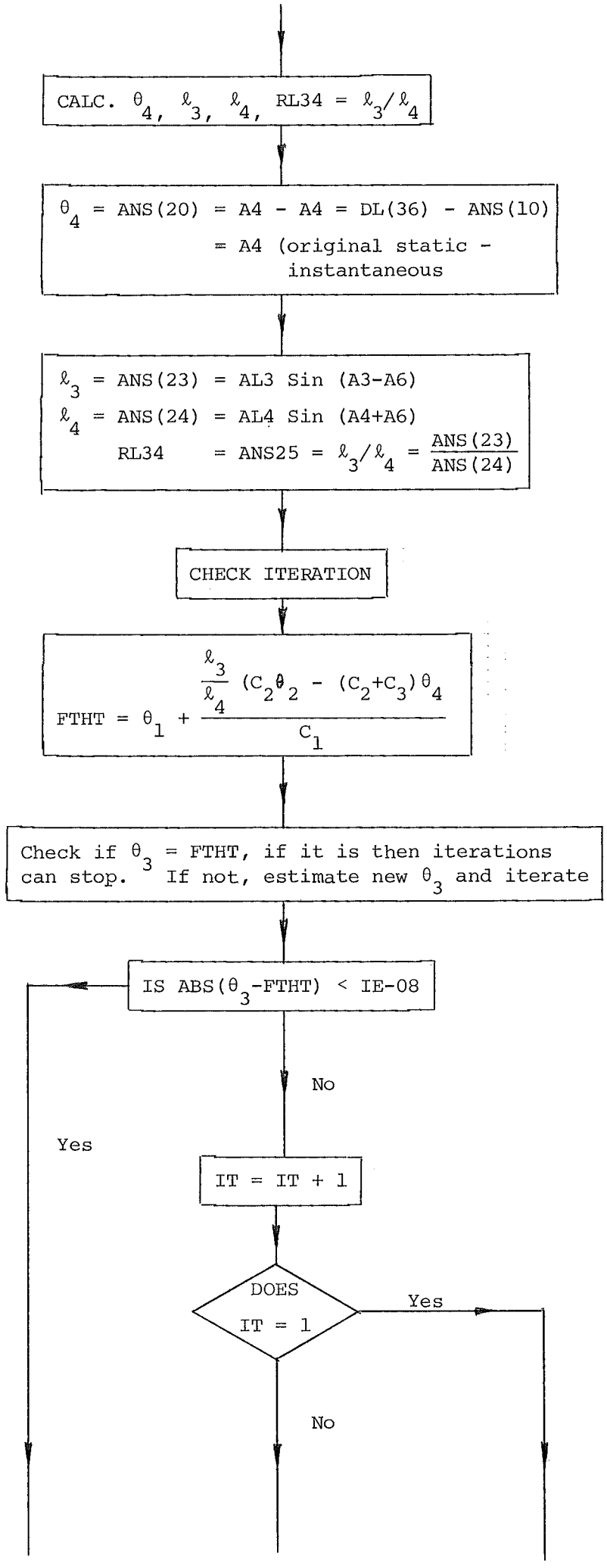
$$\text{ABS}(f(\theta_n) - \theta_n) < 10^{-8}$$

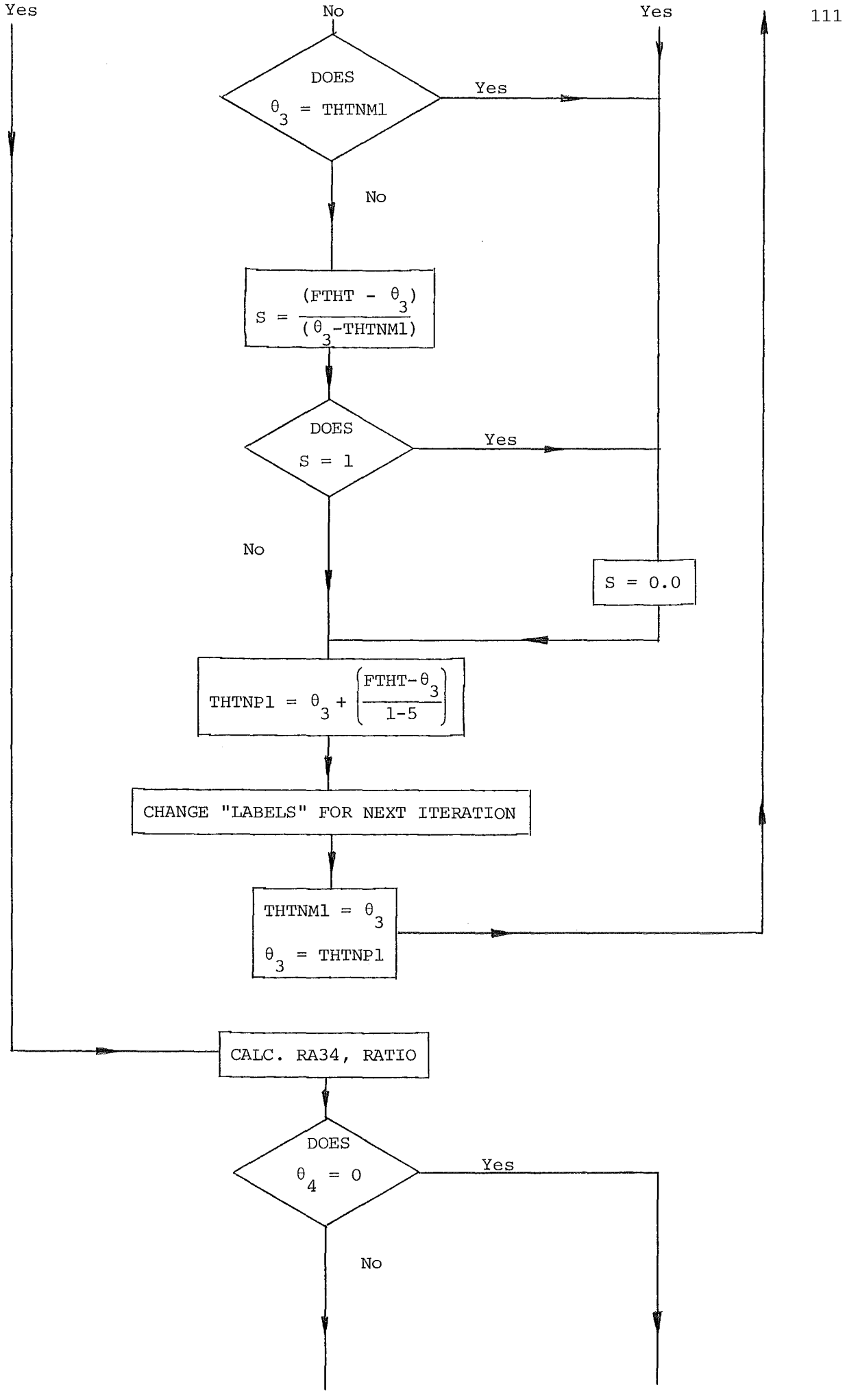
The subroutine counts and prints out the number of iterations ITS for each bump/rebound permutation specified.

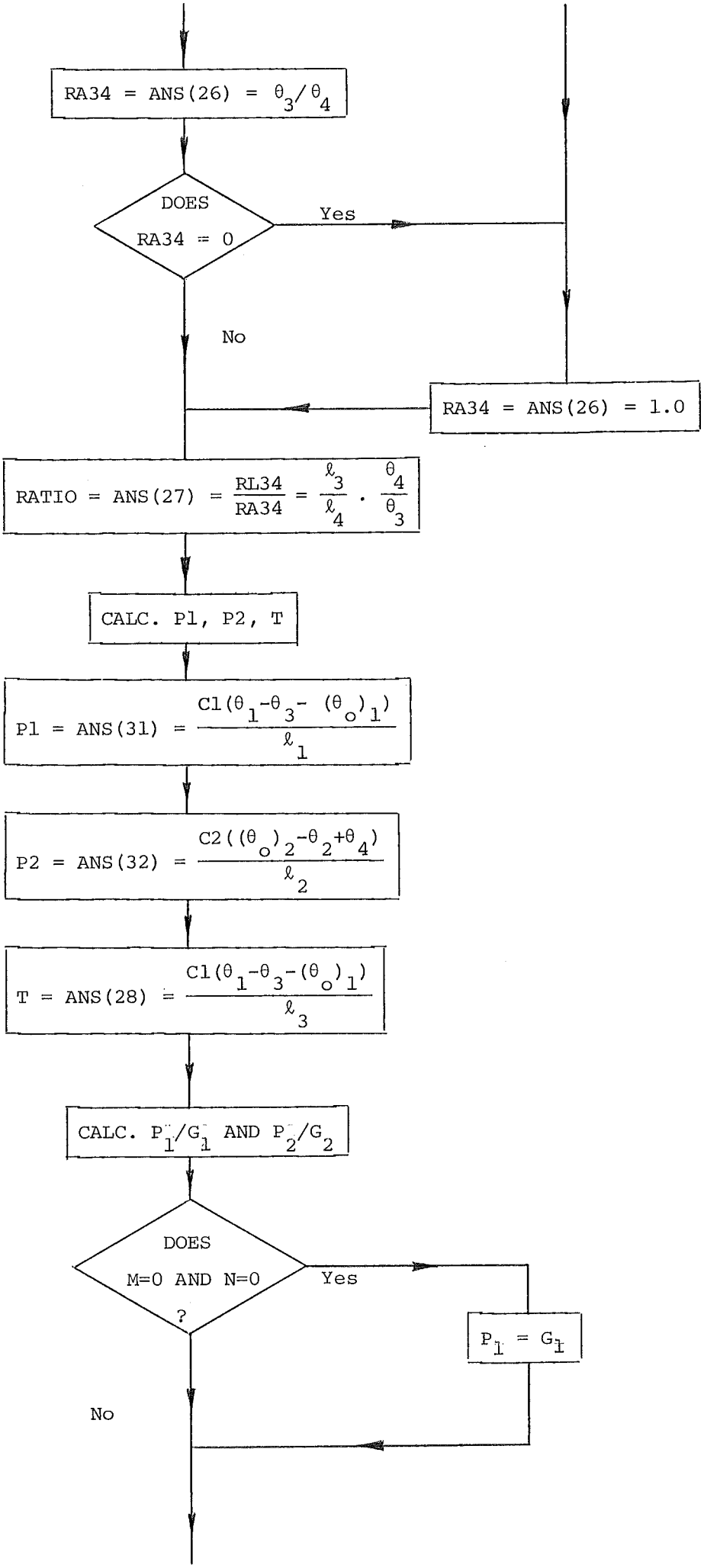
FLOWCHART

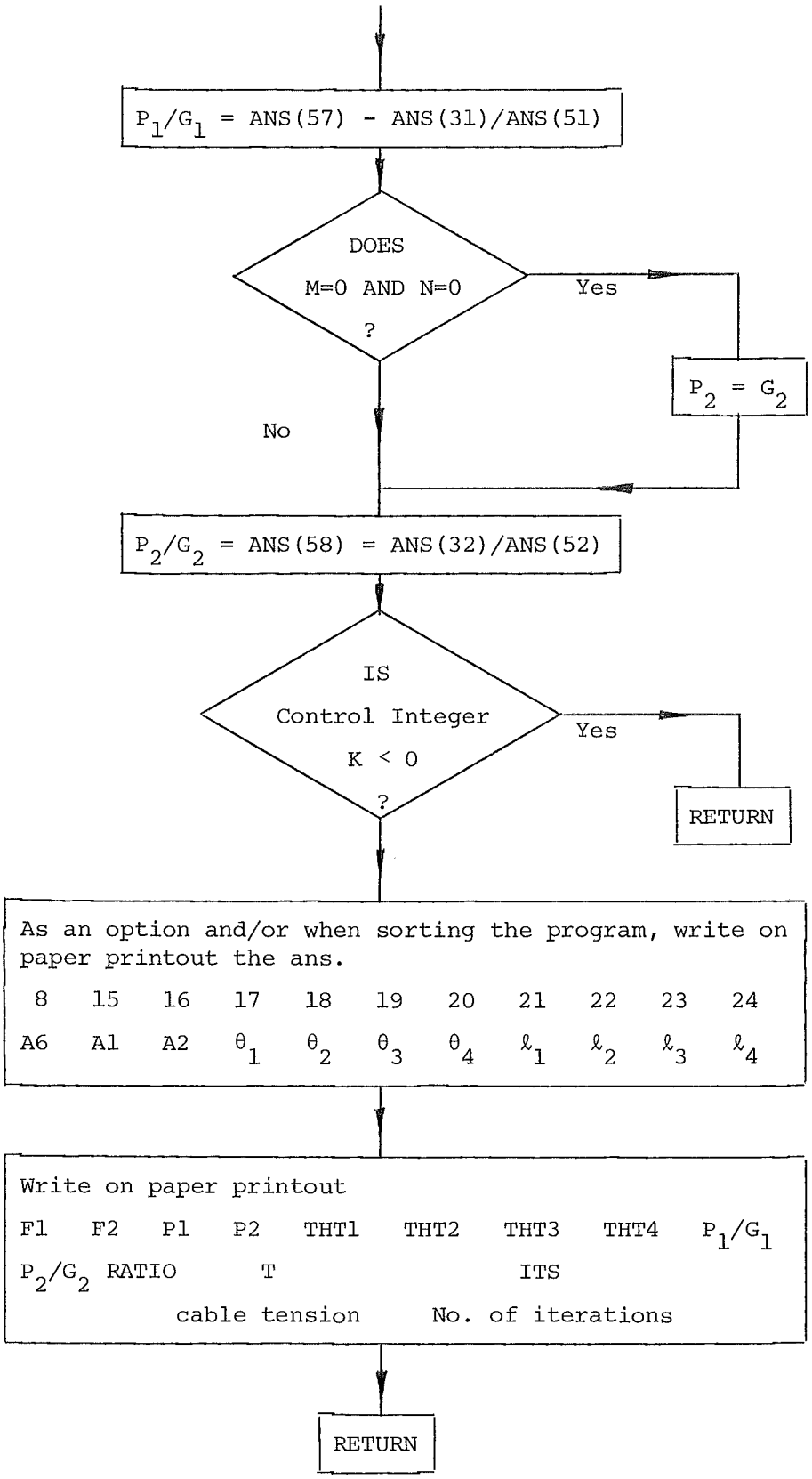

SUBROUTINE DBUMP(M,N,DL,ANS)
CALCULATES BUMP DETAILS
CALL * M FOR FRONT BUMP/REBOUND*N FOR REAR BUMP/REBOUND
CALC. Fl & F2
Fl = ANS(29) = M/1000 to convert from
F2 = ANS(30) = N/1000 mm to m for Fl & F2
CALC. THETAl, Al, Ll
θ1 = ANS(17) = ARSIN(SIN(Al) + Fl/ALl) - Al
Note Al = DL(33)
DOES
M = 0
Yes
θ1 = ANS(17) = 0
No
Al = ANS(15) = instantaneous Al, which is different from Al = DL(33) = original static cond. Al which still exists in DL(33).
Al = ANS(15) = Al + θ1 (i.e. = DL(33) + θ1
Ll = ANS(21) = ALl Cos(Al). [i.e. ALl Cos(ANS(15))]


CALC. THETA2, A2,L2
θ2 = ANS(18) = ARSIN(sin(A2) - F2/AL2) - AL
Note: A2 = DL(34)
DOES
N = 0
Yes
θ2 = ANS(18) = 0
No
A2 = ANS(16) = instantaneous A2, which is different from A2 = DL(34) = original static condition A2, which still exists in DL(34).
A2 = ANS(16) = A1 + θ2 (i.e. DL(34) + θ2)
L2 = ANS(22) = AL2 cos(A1) [i.e. AL2 cos(ANS(16))]
CALC. THETA3 BY ITERATION
SET VALUES BEFORE ITERATING
First guess
θ3 = ANS(19) = (θ1 + θ2)/3
IT = 0
A3 = ANS(9) = A3 + θ3
CALL FBLANG (910,DL,ANS)
to calculate A4 and A6


CALC. θ_4, ℓ_3, ℓ_4, RL34 = ℓ_3/ℓ_4
θ_4 = ANS(20) = A4 - A4 = DL(36) - ANS(10)
= A4 (original static - instantaneous
ℓ_3 = ANS(23) = AL3 Sin (A3-A6)
ℓ_4 = ANS(24) = AL4 Sin (A4+A6)
RL34 = ANS25 = ℓ_3/ℓ_4 = ANS(23)/ANS(24)
CHECK ITERATION
FTHT = $\theta_1 + \frac{\frac{\ell_3}{\ell_4}(C_2\theta_2 - (C_2+C_3)\theta_4}{C_1}$
Check if θ_3 = FTHT, if it is then iterations can stop. If not, estimate new θ_3 and iterate
IS ABS(θ_3-FTHT) < IE-08
No
Yes
IT = IT + 1
DOES IT = 1
Yes
No


Yes
No
Yes
DOES
θ_3 = THTNM1
Yes
No
$S = \frac{(FTHT - \theta_3)}{(\theta_3 - THTNM1)}$
DOES
S = 1
Yes
No
S = 0.0
$THTNP1 = \theta_3 + \left(\frac{FTHT - \theta_3}{1-5}\right)$
CHANGE "LABELS" FOR NEXT ITERATION
THTNM1 = θ_3
θ_3 = THTNP1
CALC. RA34, RATIO
DOES
θ_4 = 0
Yes
No


RA34 = ANS(26) = θ_3/θ_4
DOES RA34 = 0
Yes
No
RA34 = ANS(26) = 1.0
RATIO = ANS(27) = $\frac{RL34}{RA34} = \frac{\ell_3}{\ell_4} \cdot \frac{\theta_4}{\theta_3}$
CALC. P1, P2, T
P1 = ANS(31) = $\frac{C1(\theta_1-\theta_3-(\theta_o)_1)}{\ell_1}$
P2 = ANS(32) = $\frac{C2((\theta_o)_2-\theta_2+\theta_4)}{\ell_2}$
T = ANS(28) = $\frac{C1(\theta_1-\theta_3-(\theta_o)_1)}{\ell_3}$
CALC. P_1/G_1 AND P_2/G_2
DOES M=0 AND N=0 ?
Yes
$P_1 = G_1$
No


P_1/G_1 = ANS(57) - ANS(31)/ANS(51)
DOES M=0 AND N=0 ?
Yes
$P_2 = G_2$
No
P_2/G_2 = ANS(58) = ANS(32)/ANS(52)
IS Control Integer K < 0 ?
Yes
RETURN
As an option and/or when sorting the program, write on paper printout the ans.
8 15 16 17 18 19 20 21 22 23 24
A6 A1 A2 θ_1 θ_2 θ_3 θ_4 ℓ_1 ℓ_2 ℓ_3 ℓ_4
Write on paper printout
F1 F2 P1 P2 THT1 THT2 THT3 THT4 P_1/G_1
P_2/G_2 RATIO T ITS
cable tension No. of iterations
RETURN

SUBROUTINE SPRATE (K, DL, ANS)

Purpose. To calculate the interconnected springrates, both wheelrates and total vehicle rates.

Input. Subroutine uses:

ANS(1) = C_1 = Front torsion bar springrate.

ANS(3) = C_2 = Rear torsion bar springrate.

ANS(5) = C_3 = Auxiliary torsion bar springrate.

ANS(21) = Effective length ℓ_1.

ANS(22) = Effective length ℓ_2.

ANS(25) = Ratio of effective lengths ℓ_3/ℓ_4.

ANS(27) = RATIO = $\ell_3\theta_4/\ell_4\theta_3$.

Output. Subroutine calculates the following:

ANS(33) = C_{11} = front wheelrate.

ANS(34) = C_{22} = rear wheelrate.

ANS(35) = C_{12} = front wheelrate due to rear.

ANS(36) = C_{21} = rear wheelrate due to front.

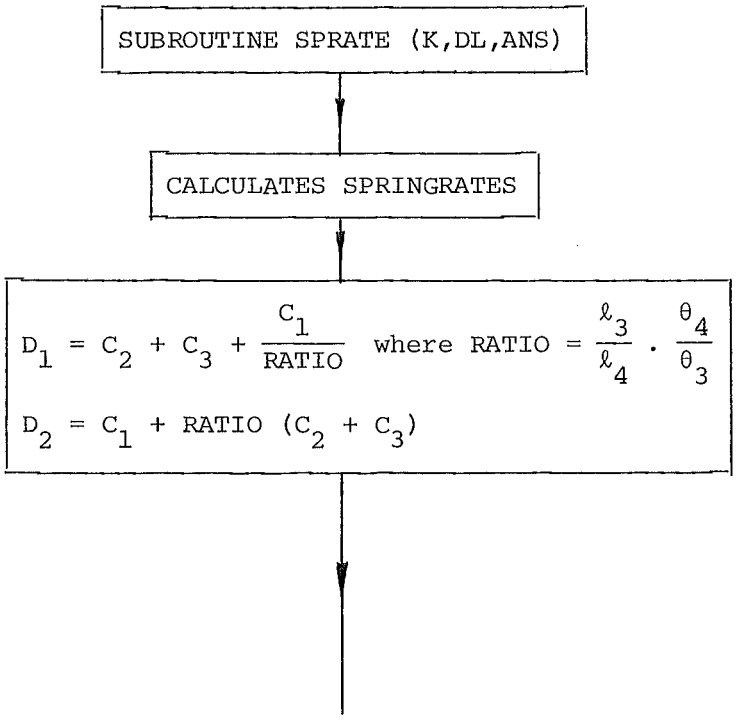
ANS(53) = 2 C_{11} = car springrate front.

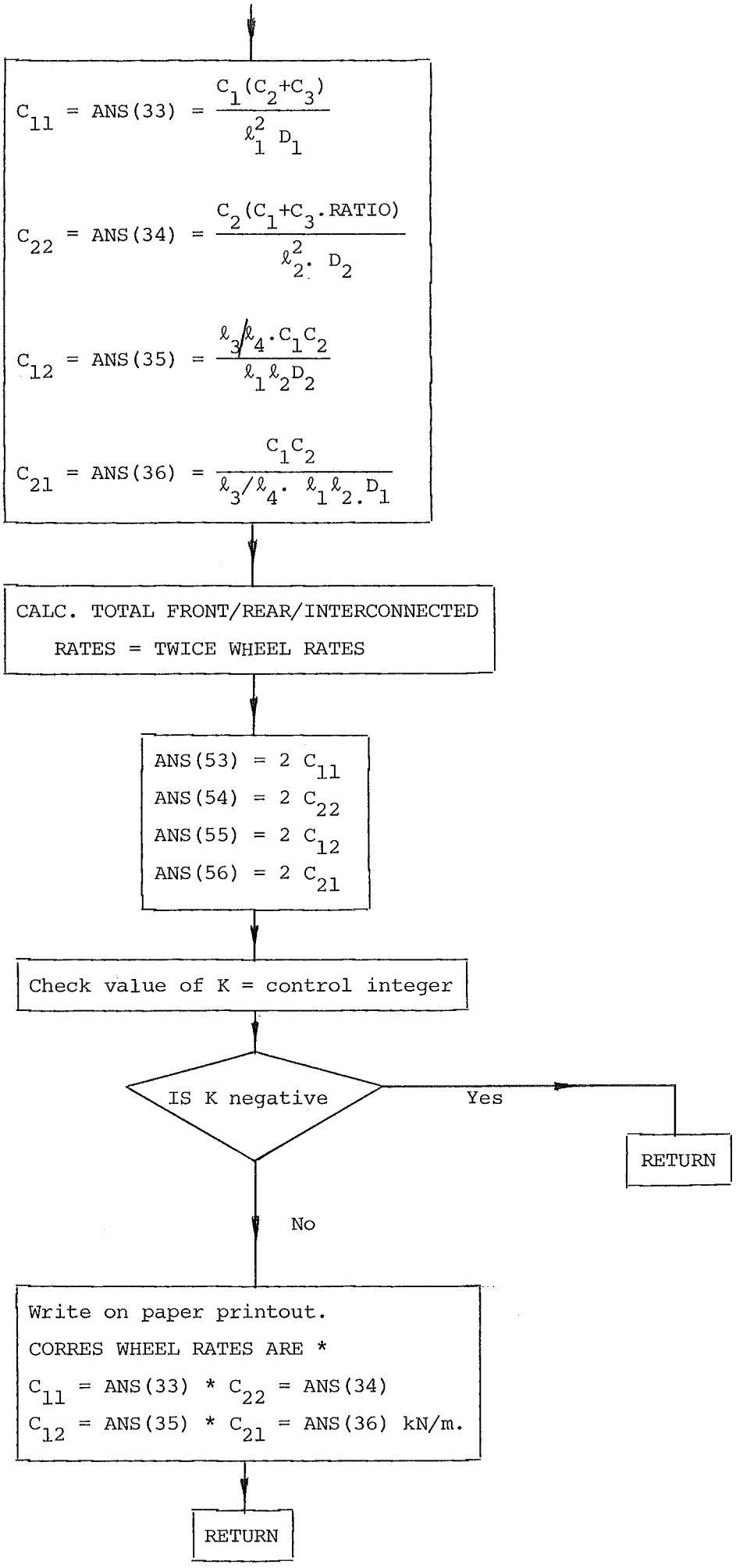
ANS(54) = 2 C_{22} = car springrate rear.

ANS(55) = 2 C_{12} = car springrate front/rear.

ANS(56) = 2 C_{21} = car springrate rear/front.

FLOWCHART




$C_{11} = ANS(33) = \frac{C_1(C_2+C_3)}{\ell_1^2 D_1}$
$C_{22} = ANS(34) = \frac{C_2(C_1+C_3 . RATIO)}{\ell_2^2 . D_2}$
$C_{12} = ANS(35) = \frac{\ell_3/\ell_4 . C_1 C_2}{\ell_1 \ell_2 D_2}$
$C_{21} = ANS(36) = \frac{C_1 C_2}{\ell_3/\ell_4 . \ell_1 \ell_2 . D_1}$
CALC. TOTAL FRONT/REAR/INTERCONNECTED
RATES = TWICE WHEEL RATES
ANS(53) = 2 C_{11}
ANS(54) = 2 C_{22}
ANS(55) = 2 C_{12}
ANS(56) = 2 C_{21}
Check value of K = control integer
IS K negative
Yes
RETURN
No
Write on paper printout.
CORRES WHEEL RATES ARE *
C_{11} = ANS(33) * C_{22} = ANS(34)
C_{12} = ANS(35) * C_{21} = ANS(36) kN/m.
RETURN

SUBROUTINE FREQ (DL, ANS)

Purpose. To calculate the bounce and pitch frequencies of the car, these frequencies being largely responsible for the comfort of the car. The partial frequencies of bounce and pitch and the coupling coefficients are also calculated.

Input. The springrates and vehicle geometry are required.

ANS(53) = 2 x C_{11}
ANS(54) = 2 x C_{22}
ANS(55) = 2 x C_{12}
ANS(56) = 2 x C_{21}

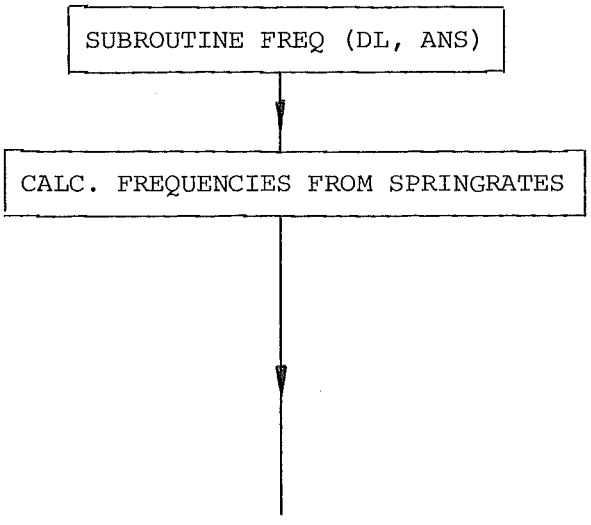
where C_{11}, C_{22}, C_{12} and C_{21} are individual wheelrates

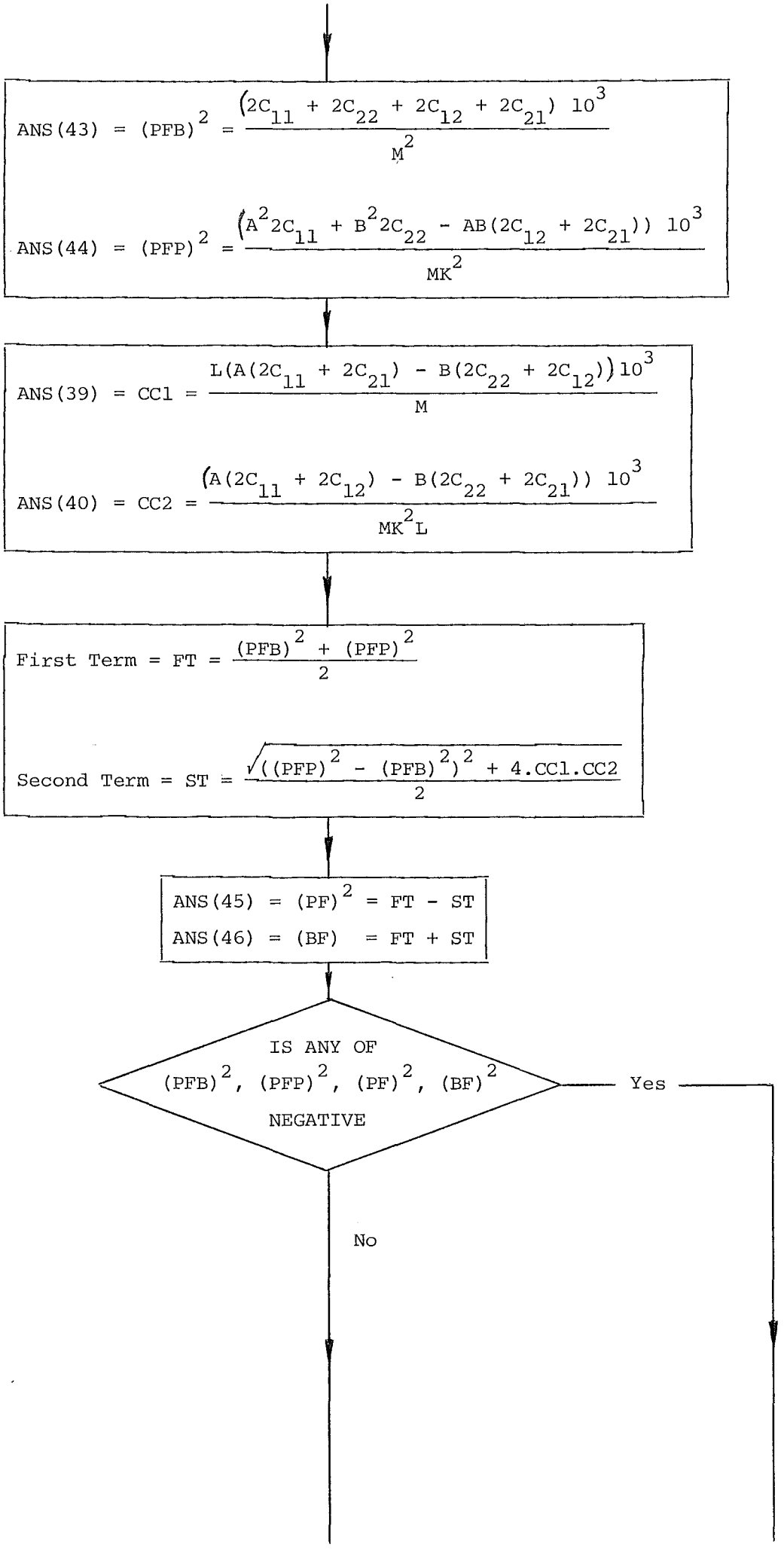
DL(41) = WB = L
DL(42) = A
DL(43) = B
DL(45) = M
DL(46) = K

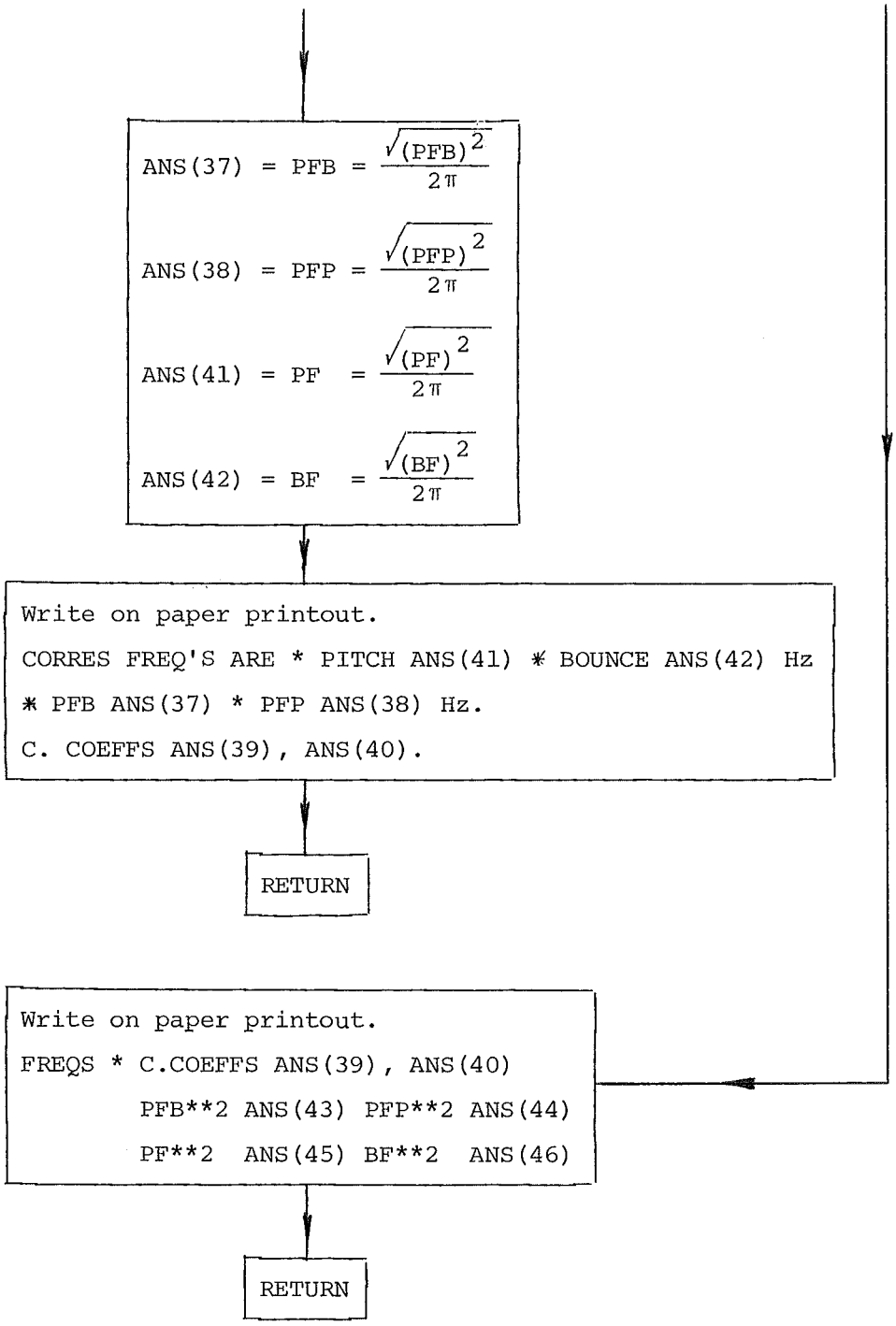
Output.

ANS(41) = PF = pitch frequency.
ANS(42) = BF = bounce frequency.
ANS(37) = PFB = partial bounce frequency.
ANS(38) = PFP = partial pitch frequency.
ANS(39) = CC1 = coupling coefficient 1.
ANS(40) = CC2 = coupling coefficient 2.

FLOWCHART




ANS(43) = (PFB)^2 = $\frac{(2C_{11} + 2C_{22} + 2C_{12} + 2C_{21})\ 10^3}{M^2}$
ANS(44) = (PFP)^2 = $\frac{(A^2 2C_{11} + B^2 2C_{22} - AB(2C_{12} + 2C_{21}))\ 10^3}{MK^2}$
ANS(39) = CC1 = $\frac{L(A(2C_{11} + 2C_{21}) - B(2C_{22} + 2C_{12}))10^3}{M}$
ANS(40) = CC2 = $\frac{(A(2C_{11} + 2C_{12}) - B(2C_{22} + 2C_{21}))\ 10^3}{MK^2L}$
First Term = FT = $\frac{(PFB)^2 + (PFP)^2}{2}$
Second Term = ST = $\frac{\sqrt{((PFP)^2 - (PFB)^2)^2 + 4.CC1.CC2}}{2}$
ANS(45) = (PF)^2 = FT - ST
ANS(46) = (BF) = FT + ST
IS ANY OF
(PFB)^2, (PFP)^2, (PF)^2, (BF)^2
NEGATIVE
Yes
No


ANS(37) = PFB = $\frac{\sqrt{(PFB)^2}}{2\pi}$
ANS(38) = PFP = $\frac{\sqrt{(PFP)^2}}{2\pi}$
ANS(41) = PF = $\frac{\sqrt{(PF)^2}}{2\pi}$
ANS(42) = BF = $\frac{\sqrt{(BF)^2}}{2\pi}$
Write on paper printout.
CORRES FREQ'S ARE * PITCH ANS(41) * BOUNCE ANS(42) Hz
* PFB ANS(37) * PFP ANS(38) Hz.
C. COEFFS ANS(39), ANS(40).
RETURN
Write on paper printout.
FREQS * C.COEFFS ANS(39), ANS(40)
PFB**2 ANS(43) PFP**2 ANS(44)
PF**2 ANS(45) BF**2 ANS(46)
RETURN

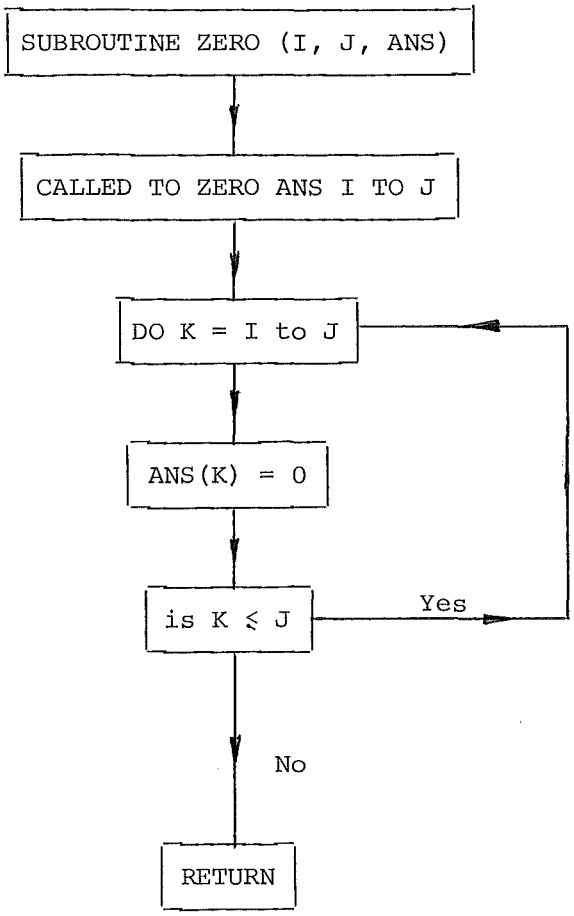
SUBROUTINE ZERO (I, J, ANS)

Purpose. To set to zero those elements in the ANS array from I to J inclusive.

Input. I = lowest ANS(I) to be set to zero.
J = highest ANS(j) to be set to zero.
ANS = Answer array.

Output. ANS(I), ANS(I+1) .... ANS(J) = 0.

FLOWCHART



SUBROUTINE STDEFL (K, DL, ANS)

Purpose. To calculate the angle of twist of the front and rear torsion bars when the vehicle is in its static condition, for subsequent total twist and stress calculations.

From the initial angle of twist is calculated the static front and rear wheel deflection and the static wheel loadings.

Input. K = control integer.

DL = Data list array.

ANS = Answers array.

This subroutine uses,

subroutine ZERO and DBUMP

DL(25) = AL1 = Front wheel to torsion bar, arm length.

DL(26) = AL2 = Rear wheel to torsion bar arm length.

DL(33) = A1 = Static angle of arm AL1.

DL(34) = A2 = Static angle of arm AL2.

DL(42) = A = A x WB is distance of C of G from front.

DL(43) = B = B x WB is distance of C of G from rear.

DL(45) = M = Vehicle mass.

ANS(1) = C_1 = Front torsion bar springrate.

ANS(3) = C_2 = Rear torsion bar springrate.

ANS(31) = P_1 = Front wheel reaction.

ANS(32) = P_2 = Rear wheel reaction.

Output. Subroutine calculates.

ANS(47) = $(\theta_o)_1$ = angle θ_1 at zero wheel loading.

ANS(48) = $(\theta_o)_2$ = angle θ_2 at zero wheel loading.

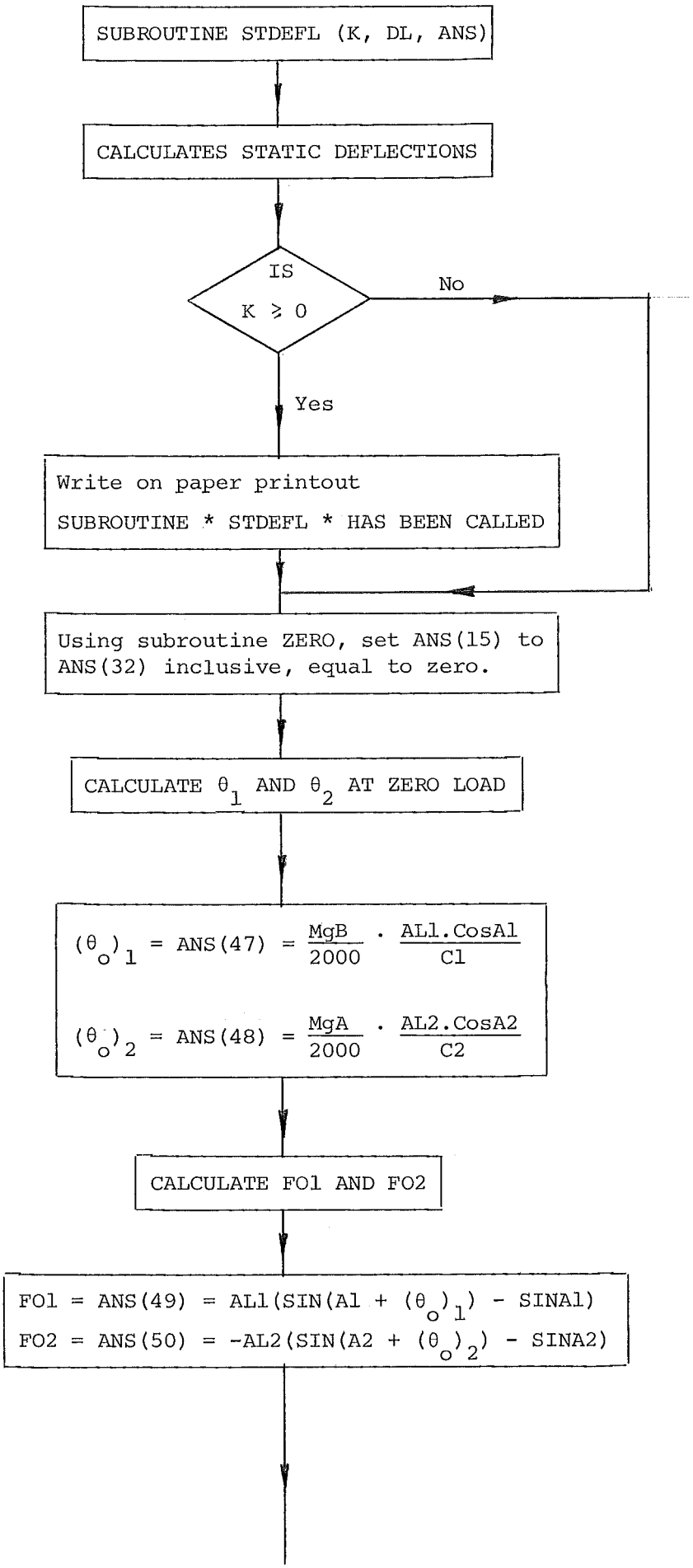
ANS(49) = F_{01} = static front wheel deflection.

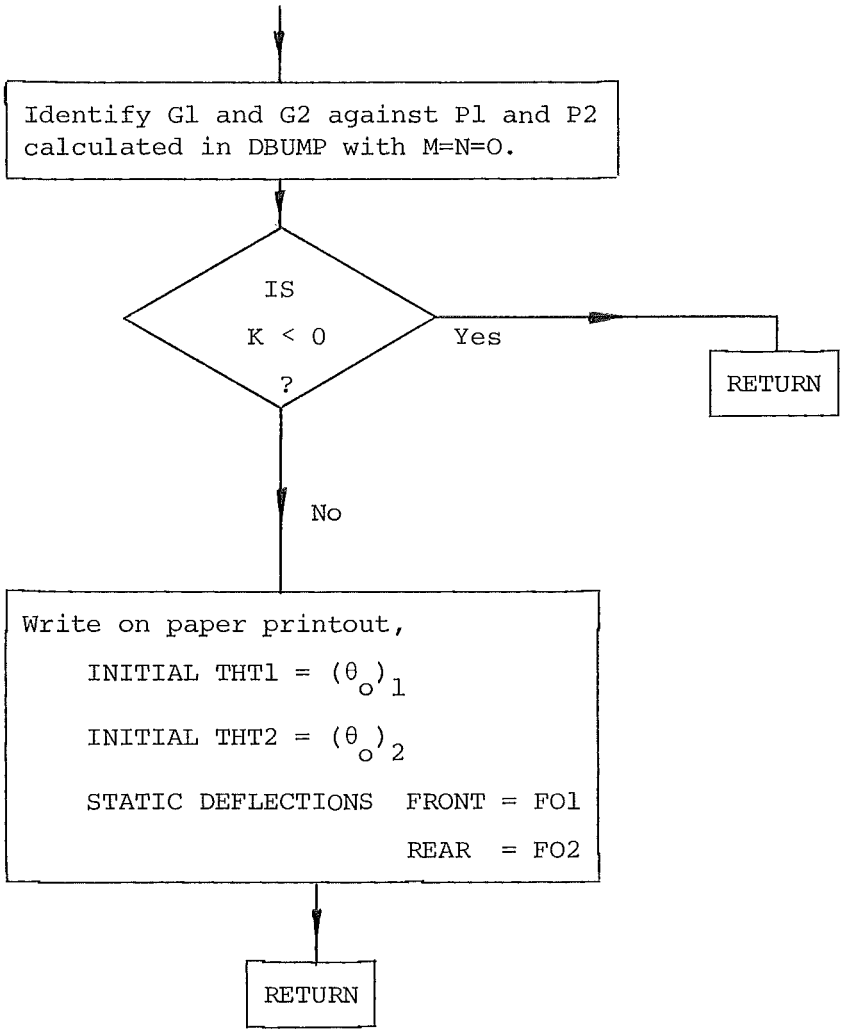
ANS(50) = F_{02} = static rear wheel deflection.

ANS(51) = G_1 = static front wheel loading.

ANS(52) = G_2 = static rear wheel loading.

FLOWCHART


SUBROUTINE STDEFL (K, DL, ANS)
CALCULATES STATIC DEFLECTIONS
IS
K ≥ 0
No
Yes
Write on paper printout
SUBROUTINE * STDEFL * HAS BEEN CALLED
Using subroutine ZERO, set ANS(15) to ANS(32) inclusive, equal to zero.
CALCULATE θ1 AND θ2 AT ZERO LOAD
(θo)1 = ANS(47) = MgB/2000 . AL1.CosA1/C1
(θo)2 = ANS(48) = MgA/2000 . AL2.CosA2/C2
CALCULATE FO1 AND FO2
FO1 = ANS(49) = AL1(SIN(A1 + (θo)1) - SINA1)
FO2 = ANS(50) = -AL2(SIN(A2 + (θo)2) - SINA2)


Identify G1 and G2 against P1 and P2 calculated in DBUMP with M=N=O.
IS
K < 0
?
Yes
RETURN
No
Write on paper printout,
INITIAL THT1 = $(\theta_o)_1$
INITIAL THT2 = $(\theta_o)_2$
STATIC DEFLECTIONS FRONT = FO1
REAR = FO2
RETURN

SUBROUTINE STRESS (K, D, A)

Purpose. To calculate the total windup angles and stresses of the front, rear and auxiliary torsionbars for design purposes.

Input. K = control integer.

D = Data List Array.

A = Answers Array.

Subroutine uses:

A(17) = θ_1 = front arm angular displacement, bump +ve.

A(19) = θ_3 = front pitch arm angular displacement.

A(47) = $(\theta_o)_1 = \theta_1$ at zero wheel loading (-ve).

A(18) = θ_2 = rear arm angular displacement, bump +ve.

A(20) = θ_4 = rear pitch arm angular displacement.

A(48) = $(\theta_o)_2 = \theta_2$ at zero wheel loading (+ve).

A(2) = Front torsion bar stressrate.

A(4) = Rear torsion bar stressrate.

A(6) = Auxiliary torsion bar stressrate.

Output. A = Answer Array.

A(59) = ϕ_1 = windup angle for torsion bar C_1 (front).

A(60) = ϕ_2 = windup angle for torsion bar C_2 (rear).

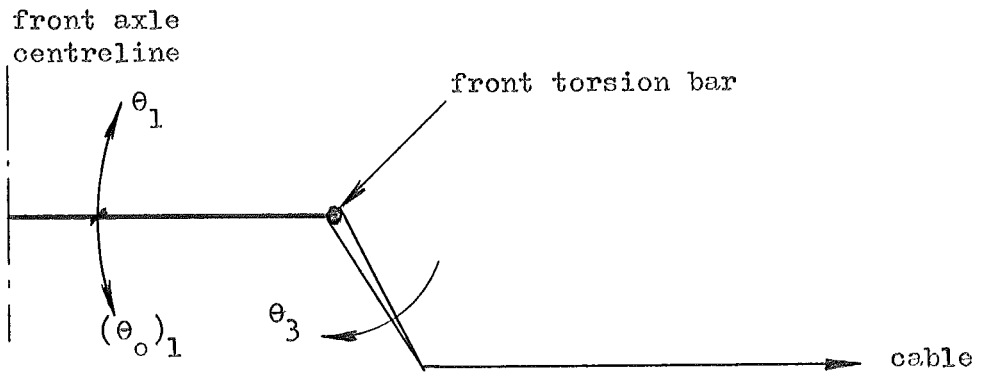
A(61) = S_1 = stress for torsion bar C_1 (front).

A(62) = S_2 = stress for torsion bar C_2 (rear).

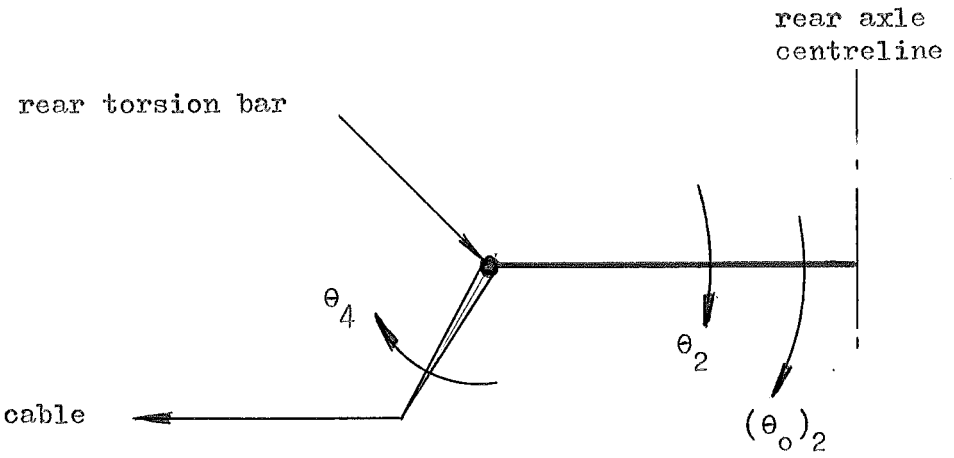
A(63) = S_3 = stress for torsion bar C_3 (auxiliary).

Description

$$A(59) = \phi_1 = \theta_1 - \theta_3 - (\theta_o)_1 \; ; \; (\theta_o)_1 \text{ is negative.}$$



A(60) = $\phi_2 = \theta_2 - \theta_4 - (\theta_o)_2$; $(\theta_o)_2$ is positive.



θ_1, θ_3 are positive for front wheel bump.

θ_2, θ_4 are negative for rear wheel bump.

$(\theta_o)_1$, $(\theta_o)_2$ correspond to zero wheel or torsion bar load, so that at static conditions when $\theta_1 = \theta_2 = 0$, $\theta_1 = -(\theta_o)_1$, is positive and $\theta_2 = -(\theta_o)_2$ is negative.

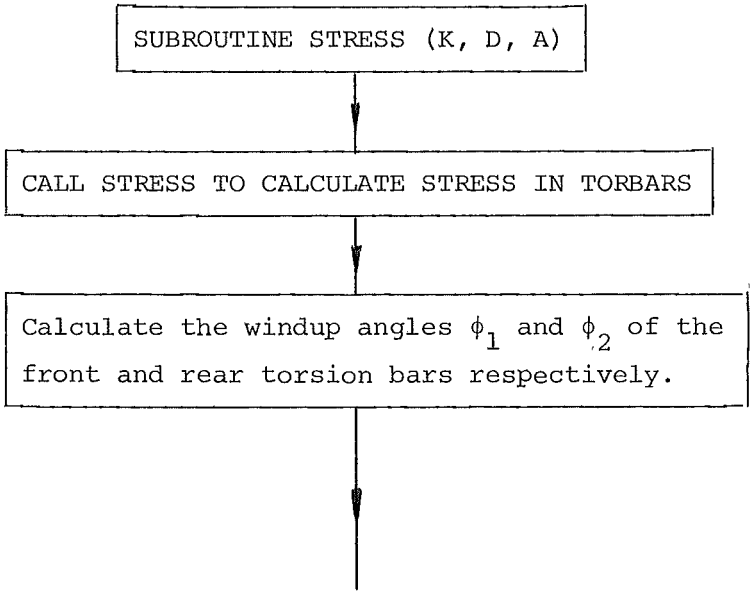
A(61) = S_1 = $(\text{stresrate})_1 \cdot \theta_1$

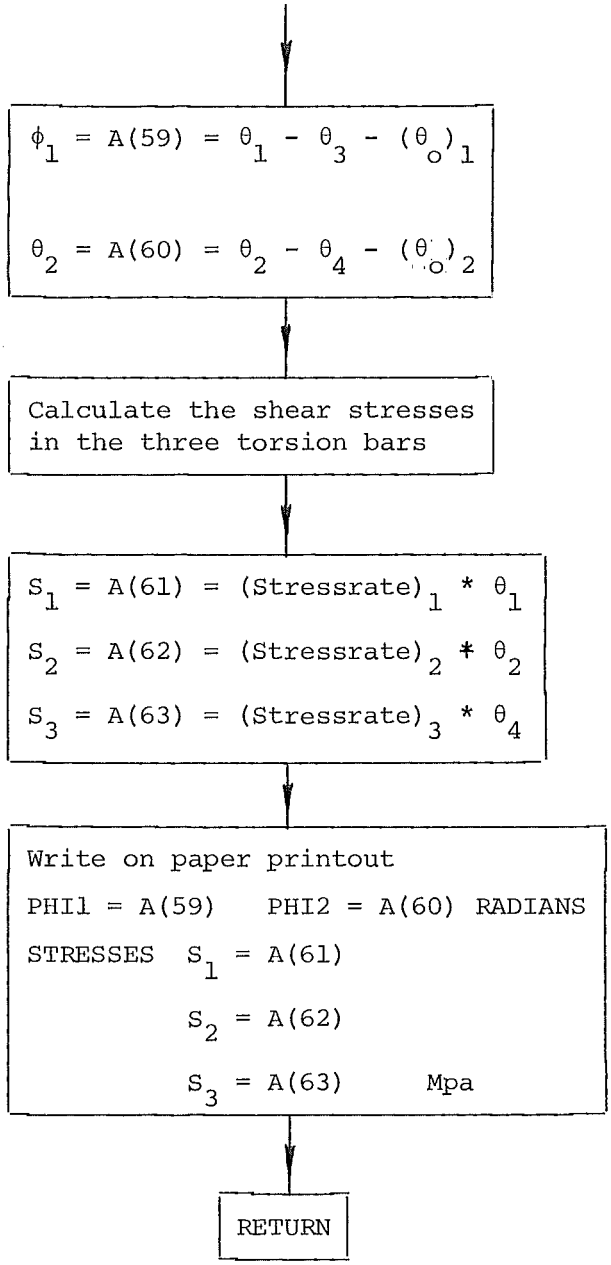
A(62) = S_2 = $(\text{stressrate})_2 \cdot \theta_2$

A(63) = S_3 = $(\text{stressrate})_3 \cdot \theta_4$

For calculating S_3, the angle of twist is θ_4, that is, it is equal to that of the rear pitch arm. The auxiliary torsion bar is connected linearly to the rear pitch arm.

FLOWCHART




ϕ_1 = A(59) = $\theta_1 - \theta_3 - (\theta_o)_1$
θ_2 = A(60) = $\theta_2 - \theta_4 - (\theta_o)_2$
Calculate the shear stresses in the three torsion bars
S_1 = A(61) = (Stressrate)$_1$ * θ_1
S_2 = A(62) = (Stressrate)$_2$ * θ_2
S_3 = A(63) = (Stressrate)$_3$ * θ_4
Write on paper printout
PHI1 = A(59) PHI2 = A(60) RADIANS
STRESSES S_1 = A(61)
S_2 = A(62)
S_3 = A(63) Mpa
RETURN

I N T E R S U S P

```
C  DATA MUST BE FOLLOWED WITH A BLANK CARD
C
      DIMENSION DL(100),ANS(100)
C
    1 CALL ZERO(1,50,ANS)
      READ(5,2)(DL(I),I=1,8)
      002:0010:1  IS THE LOCATION FOR EXCEPTIONAL ACTION ON THE I/O STATE
      IF(DL(1).EQ.0)GO TO 99
      READ(5,2)(DL(I),I=9,64)
      002:0021:1  IS THE LOCATION FOR EXCEPTIONAL ACTION ON THE I/O STATE
      CALL SORTDT(DL)
      CALL TORBAR(DL,ANS)
      CALL ARMLTH(DL,ANS)
      CALL STDEFL(0,DL,ANS)
      CALL BUMP(100,100,50,DL,ANS)
   99 CONTINUE
      END



      SUBROUTINE SORTDT(DL)
C  TO SORT AND WRITE DATA
      DIMENSION DL(1)
      WRITE(6,200)
  200 FORMAT( //' PROGRAM CONTROL INTEGERS')
      WRITE(6,100)(DL(I),I=1,8)
  100 FORMAT( /,' ITF =',I5,'  FRONT TORSION BAR (1=CIRCULAR,2=RECTANGUL
     TAR)',/,' ITR =',I5,'  REAR TORSION BAR (1=CIRCULAR,2=RECTANGULAR)'
     T/,' IAL =',I5,'  ARM LENGTH DATA',
     T/,' IAD =',I5,'  INITIAL ANGLE DATA',
     T/,' IVG =',I5,'  VEHICLE GEOMETRY DATA',
     T/,6H LBA =,I5,30H  LOAD-BRAKE-ACCELERATION DATA,
     T/,6H ITA =,I5,44H  AUX TORSION BAR (1=CIRCULAR,2=RECTANGULAR),
     T/,(I5))
      IF (DL(1).EQ.1)WRITE(6,111)DL(1),(DL(I),I=9,16)
  111 FORMAT( //,' FRONT TORSION BAR * * CIRCULAR * * ITF =',I5,
     T/,6X,'G',9X,'L',9X,'D',/,1X,E10.5,7F10.4)
      IF (DL(1).EQ.2)WRITE(6,112)DL(1),(DL(I),I=9,16)
  112 FORMAT( //,' FRONT TORSION BAR * * RECTANGULAR * * ITF =',I5,
     T/,6X,'G',9X,'L',9X,'A',9X,'B',8X,'N2',8X,'N3',/,1X,E10.5,7F10.4)
      IF (DL(2).EQ.1)WRITE(6,121)DL(2),(DL(I),I=17,24)
  121 FORMAT( //,' REAR TORSION BAR * * CIRCULAR * * ITR = ',I5,
     T/,6X,'G',9X,'L',9X,'D',/,1X,E10.5,7F10.4)
      IF (DL(2).EQ.2)WRITE(6,122)DL(2),(DL(I),I=17,24)
  122 FORMAT( //,' REAR TORSION BAR * * RECTANGULAR * * ITR =',I5,
     T/,6X,'G',9X,'L',9X,'A',9X,'B',8X,'N2',8X,'N3',/,1X,E10.5,7F10.4)
      IF (DL(3).EQ.1)WRITE(6,131)DL(3),(DL(I),I=25,32)
  131 FORMAT( //,' ARM LENGTH DATA * * IAL =',I5,
     T/,4X,'AL1',7X,'AL2',7X,'AL3',7X,'AL4',7X,'AL5',7X,'AL6',8X,'L3',
     T/,8F10.4)
      IF (DL(4).EQ.1)WRITE(6,141)DL(4),(DL(I),I=33,40)
  141 FORMAT( //,' INITIAL ANGLE DATA * * IAD =',I5,
     T/,5X,'A1',8X,'A2',8X,'A3',8X,'A4',8X,'A5',8X,'A6',/,8F10.4)
      IF(DL(5).EQ.1)WRITE(6,151)DL(5),(DL(I),I=41,48)
  151 FORMAT( //,' VEHICLE GEOMETRY DATA * * IVG =',I5,
     T/,5X,'WB',9X,'A',9X,'B',9X,'H',9X,'M',9X,'K',/,8F10.4)
      IF(DL(42)+DL(43).NE.1.0)WRITE(6,159)
  159 FORMAT( 3(/,33H * * A+B DOES NOT EQUAL UNITY * *))
      IF(DL(6).EQ.1)WRITE(6,161)DL(6),(DL(I),I=49,56)
  161 FORMAT( //,39H LOAD-BRAKE-ACCELERATION DATA * * LBA =,I5,
     T/,4X,3HDFL,7X,3HMFL,7X,3HDRL,7X,3HMRL,8X,2HDB,8X,2HMB,8X,2HDA,
     T8X,2HMA,/,8F10.4)
      IF(DL(7).EQ.1)WRITE(6,171)DL(7),(DL(I),I=57,64)
  171 FORMAT( //,46H AUXILIARY TORSION BAR * * CIRCULAR * * ITA =,I5,
     T/,6X,'G',9X,'L',9X,'D',/,1X,E10.5,7F10.4)
      IF(DL(7).EQ.2)WRITE(6,172)DL(7),(DL(I),I=57,64)
  172 FORMAT( //,48H AUXILIARY TORSION BAR * * RECTANGULAR * * ITA =,I5,
     T/,6X,'G',9X,'L',9X,'A',9X,'B',8X,'N2',8X,'N3',/,1X,E10.5,7F10.4)
      RETURN
      END
```

```
      SUBROUTINE TORBAR(DL,ANS)
C TO CALCULATE * * TORSIONAL RATE * * STRESS RATE * *
      DIMENSION DL(1),ANS(1)
C
C * FRONT  TORATE C1   STRESSRATE  * * REAR  TORATE C2   STRESS RATE *
C           ANS(1)        ANS(2)                ANS(3)     ANS(4)
C
      IF (DL(1).EQ.0) GO TO 2
      IF (DL(1).EQ.2) GO TO 1
C CIRCULAR
      ANS(1)=3.14159265358*(DL(11)**4)*DL(9)/(32*DL(10)*1.E+03)
      ANS(2)=DL(11)*DL(9)/(2*DL(10)*1.E+06)
      GO TO 2
    1 CONTINUE
C RECTANGULAR
      ANS(1)=DL(14)*DL(11)*DL(11)*DL(11)*DL(12)*DL(9)/(DL(10)*1.E+03)
      ANS(2)=DL(14)*DL(11)*DL(9)/(DL(13)*DL(10)*1.E+06)
    2 CONTINUE
      IF (DL(2).EQ.0) GO TO 4
      IF (DL(2).EQ.2) GO TO 3
C CIRCULAR
      ANS(3)=3.14159265358*(DL(19)**4)*DL(17)/(32*DL(18)*1.E+03)
      ANS(4)=DL(19)*DL(17)/(2*DL(18)*1.E+06)
      GO TO 4
    3 CONTINUE
C RECTANGULAR
      ANS(3)=DL(22)*DL(19)*DL(19)*DL(19)*DL(20)*DL(17)/(DL(18)*1.E+03)
      ANS(4)=DL(22)*DL(19)*DL(17)/(DL(21)*DL(18)*1.E+06)
    4 CONTINUE
      IF(DL(7).EQ.0)GO TO 6
      IF(DL(7).EQ.2)GO TO 5
C  CIRCULAR
      ANS(5)=3.14159265358*(DL(59)**4)*DL(57)/(32*DL(58)*1.E+03)
      ANS(6)=DL(59)*DL(57)/(2*DL(58)*1.E+06)
      GO TO 6
    5 CONTINUE
C  RECTANGULAR
      ANS(5)=DL(62)*DL(59)*DL(59)*DL(59)*DL(60)*DL(57)/(DL(58)*1.E+03)
      ANS(6)=DL(62)*DL(59)*DL(57)/(DL(61)*DL(58)*1.E+06)
    6 CONTINUE
      WRITE(6,100)(ANS(I),I=1,6)
  100 FORMAT( /,' EACH FRONT TORSION BAR * TORSION RATE C1 =',F10.3,
     T' KNM/RAD',/,23X,' * STRESSRATE      = ',F10.3,' MPA/RAD',
     T//,' EACH REAR TORSION BAR  * TORSION RATE C2 =',F10.3,' KNM/RAD',
     T/,23X,' * STRESS RATE       =',F10.3,' MPA/RAD',
     T//,' EACH AUXILIARY TOR BAR * TORSION RATE C3 =',F10.3,' KNM/RAD',
     T/,23X,' * STRESSRATE        =',F10.3,' MPA/RAD')
   99 CONTINUE
      RETURN
      END


      SUBROUTINE FRLANG(K,D,A)
C SUBROUTINE TO FIND THE FOUR-BAR-LINKAGE ANGLES GIVEN ONE ANGLE AND THE
C K IS A CONTROL INTEGER
      DIMENSION D(1),A(1)
C DATA LIST   27    28    29    30    35    36    38
CORRES TO     AL3   AL4   AL5   AL6   A3    A4    A6
      TOP=D(29)*D(29)+D(27)*D(27)-2*D(29)*D(27)*COS(A(9))
      ENDTM=ATAN(D(27)*SIN(A(9))/(D(29)-D(27)*COS(A(9))))
CABLE ANGLE A6 = PHI = ANS(8)
      A(8)=ARCOS((TOP+D(30)*D(30)-D(28)*D(28))/(2*D(30)*SQRT(TOP)))
     T-ENDTM
CALC ANGLE4 = A(10)
      A(10)=ARCOS((TOP+D(28)*D(28)-D(30)*D(30))/(2*D(28)*SQRT(TOP)))
     T+ENDTM
   99 RETURN
      END
```

```
      SUBROUTINE ARMLTH(DL,ANS)
CALL ARMLTH TO CALC PITCH ARM LENGTHS * CABLE LENGTH * CABLE ANGLE *
CIVEN INITIAL PITCH ANGLES AND PITCH ARM OFFSET(3)
C DATA LIST   25    26    27    28    29    30    35    36    38    42    43
CORRES TO    AL1   AL2   AL3   AL4   AL5   AL6    A3    A4    A6     A     B
C     ANS                   7     9    10
CORRES                    RL34  ANG3  ANG4
      DIMENSION DL(1),ANS(1)
      WRITE(6,104)
  104 FORMAT(//,38H SUBROUTINE * ARMLTH * HAS BEEN CALLED)
      ANS(7)=DL(43)*DL(25)*COS(DL(33))/(DL(42)*DL(26)*COS(DL(34)))
CALC L4 FROM L3=DL(31)=ANS(23)
      ANS(23)=DL(31)
      ANS(24)=ANS(23)/ANS(7)
CALC A6
      DL(38)=ARSIN((ANS(24)-ANS(23))/DL(29))
CALC AL3 AL4 AL6
      DL(27)=ANS(23)/SIN(DL(35)-DL(38))
      DL(28)=ANS(24)/SIN(DL(36)+DL(38))
      DL(30)=(DL(29)-DL(27)*COS(DL(35))-DL(28)*COS(DL(36)))/COS(DL(38))
      WRITE(6,100)ANS(24),DL(27),DL(28),DL(30),DL(38)
  100 FORMAT( /,22H EFFECTIVE LENGTH L4 =,F10.4,/,
     T20H ARM LENGTHS * AL3 =,F10.4,11H  AND AL4 =,F10.4,7H METRES,/,
     T19H CABLE LENGTH AL6 =,F10.4,29H AND INITIAL CABLE ANGLE A6 =,
     TF10.4,8H RADIANS)
CALCULATE LOCKED PITCH ANGLES * UNEQUAL PITCH ARMS
CALC FRONT BUMP          11      12   * REAR BUMP     13      14
CORRES LOCKED ANGLE FRONT    REAR                  FRONT     REAR
      IF(DL(35).EQ.1.5708.AND.DL(36).EQ.1.5708)RETURN
      IF(DL(35).EQ.3.1415926536/2..AND.DL(36).EQ.3.14159265 36/2.) RETURN
      S=(DL(27)+DL(28)+DL(29)+DL(30))/2
       ANS(11)=2*ARSIN(SQRT((S-DL(27))*(S-DL(29))/(DL(27)*DL(29))))
      ANS(12)=2*ARCOS(SQRT(S*(S-DL(27))/(DL(29)*(DL(30)+DL(28)))))
      ANS(13)=2*ARCOS(SQRT(S*(S-DL(28))/(DL(29)*(DL(30)+DL(27)))))
      ANS(14)=2*ARSIN(SQRT((S-DL(28))*(S-DL(29))/(DL(28)*DL(29))))
      WRITE(6,105)(ANS(I),I=11,14)
  105 FORMAT( /,47H FRONT BUMP * LOCKED PITCH ANGLES ARE * FRONT =,
     TF10.4,11H AND REAR =,F10.4,8H RADIANS,/,46H REAR BUMP * LOCKED PIT
     TCH ANGLES ARE * FRONT =,F10.4,11H AND REAR =,F10.4,8H RADIANS)
      RETURN
      END



      SUBROUTINE BUMP(I,J,K,DL,ANS)
CALC BUMP CONFIGURATION, REACTIONS, SPRINGRATES, FREQUENCIES
      DIMENSION DL(1),ANS(1)
CODE ANSWERS  8     9    10    15    16    17    18    19    20    21    22    23
CORRES TO    A6    A3    A4    A1    A2  THT1  THT2  THT3  THT4    L1    L2    L3
C
CODE ANSWERS 24    25    26    27    28    29    30    31    32    33    34    35
CORRES TO    L4  RL34  RA34 RATIO     T    F1    F2    P1    P2   C11   C22   C12
C
CODE ANSWERS 36    37    38    39    40    41    42    47    48    49    50
CORRES TO   C21   PFB   PFP   CC1   CC2    PF    BF  ITH1  ITH2   F01   F02
C
CODE ANSWERS 51    52    53    54    55    56    57    58
CORRES TO    G1    G2  2C11  2C22  2C12  2C21  P/G1  P/G2
C
      WRITE(6,103)
  103 FORMAT( /,36H SUBROUTINE * BUMP * HAS BEEN CALLED)
CONTROL INTEGERS I AND J ARE BOUNCE/REBOUND FRONT AND REAR
CONTROL INTEGER K FOR STEPS OF BOUNCE/REBOUND
CALL * M FOR FRONT BUMP/REBOUND * N FOR REAR BUMP/REBOUND
CALCULATE FOR BOUNCE AND REBOUND
      IF(I.EQ.0)GO TO 1
      CALL ZERO(15,32,ANS)
      DO 1 M=-J,I,K
      DO 1 N=-J,I,K
      CALL DBUMP(M,N,DL,ANS)
      CALL SPRATE(K,DL,ANS)
      CALL FREQ(DL,ANS)
      CALL STRESS(0,DL,ANS)
    1 CONTINUE
      RETURN
      END
```

```
      SUBROUTINE DBUMP(M,N,DL,ANS)
CALCULATES BUMP DETAILS
CALL * M FOR FRONT BUMP/REBOUND * N FOR REAR BUMP/REBOUND
      DIMENSION DL(1),ANS(1)
CALC F1 AND F2
      ANS(29)=M/1000.
      ANS(30)=N/1000.
CALC THETA1, A1, L1
      ANS(17)=ARSIN(SIN(DL(33))+ANS(29)/DL(25))-DL(33)
      IF(M.EQ.0)ANS(17)=0.0
      ANS(15)=DL(33)+ANS(17)
      ANS(21)=DL(25)*COS(ANS(15))
CALC THETA2, A2, L2
      ANS(18)=ARSIN(SIN(DL(34))-ANS(30)/DL(26))-DL(34)
      IF(N.EQ.0)ANS(18)=0.0
      ANS(16)=DL(34)+ANS(18)
      ANS(22)=DL(26)*COS(ANS(16))
CALC THETA3 BY ITERATION
C   SET VALUES BEFORE ITERATING
      ANS(19)=(ANS(17)+ANS(18))/3.0
      IT=0
    1 CONTINUE
      ANS(9)=DL(35)+ANS(19)
      CALL FBLANG(9,10,DL,ANS)
CALC THETA4, L3, L4, RL34
      ANS(20)=DL(36)-ANS(10)
      ANS(23)=DL(27)*SIN(ANS(9)-ANS(8))
      ANS(24)=DL(28)*SIN(ANS(10)+ANS(8))
      ANS(25)=ANS(23)/ANS(24)
CHECK ITERATION
      FTHT= ANS(17)+ANS(25)*(ANS(3)*ANS(18)-(ANS(3)+ANS(5))*ANS(20))/
     TANS(1)
      IF(ABS(ANS(10)-FTHT).LT.1E-08)GO TO 2
      IT=IT+1
      IF(IT.EQ.1)GO TO 5
      IF (ANS(19).EQ.THTNM1)GO TO 5
      S=(FTHT-ANS(19))/(ANS(19)-THTNM1)
      IF(S.EQ.1.0)GO TO 5
      GO TO 6
    5 S=0.0
    6 CONTINUE
      THTNP1=ANS(10)+(FTHT-ANS(19))/(1.0-S)
CHANGE FOR NEW ITERATIONS
      THTNM1=ANS(10)
       ANS(19)=THTNP1
      GO TO 1
    2 CONTINUE
CALC RA34,RATIO
      IF(ANS(20))3,7,3
    7 ANS(26)=1.0
      GO TO 4
    3 CONTINUE
      ANS(26)=ANS(19)/ANS(20)
      IF(ANS(26).EQ.0.0)ANS(26)=1.0
    4 CONTINUE
      ANS(27)=ANS(25)/ANS(26)
CALC P1,P2,T
      ANS(31)=ANS(1)*(ANS(17)-ANS(19)-ANS(47))/ANS(21)
      ANS(32)=ANS(3)*(ANS(48)-ANS(18)+ANS(20))/ANS(22)
      ANS(28)=ANS(1)*(ANS(17)-ANS(19)-ANS(47))/ANS(23)
CALC P1/G1 AND P2/G2
      IF(M.EQ.0.AND.N.EQ.0)ANS(51)=ANS(31)
      ANS(57)=ANS(31)/ANS(51)
      IF(M.EQ.0.AND.N.EQ.0)ANS(52)=ANS(32)
      ANS(58)=ANS(32)/ANS(52)
      IF(K.LT.0)RETURN
C     WRITE(6,100)8,(I,I=15,24),ANS(8),(ANS(I),I=15,24)
  100 FORMAT( 1I10,/,11F10.4)
      WRITE(6,101)(ANS(I),I=29,32),(ANS(I),I=17,20),
     TANS(57),ANS(58),ANS(27),ANS(28),IT
  101 FORMAT( /,2H *,6X,2HF1,8X,2HF2,8X,2HP1,8X,2HP2,6X,4HTHT1,6X,4HTHT2
     T,6X,4HTHT3,6X,4HTHT4,6X,4HP/G1,6X,4HP/G2,5X,5HRATIO,9X,1HT,7X,3HIT
     TS,/,12F10.4,I10)
      RETURN
      END
```

```
      SUBROUTINE SPRATE(K,DL,ANS)
CALCULATES SPRINGRATES
      DIMENSION DL(1),ANS(1)
      D1=ANS(3)+ANS(5)+ANS(1)/ANS(27)
      D2=ANS(1)+ANS(27)*(ANS(3)+ANS(5))
C11
      ANS(33)=ANS(1)*(ANS(3)+ANS(5))/(ANS(21)*ANS(21)*D1)
C22
      ANS(34)=ANS(3)*(ANS(1)+ANS(5)*ANS(27))/(ANS(22)*ANS(22)*D2)
C12
      ANS(35)=ANS(25)*ANS(1)*ANS(3)/(ANS(21)*ANS(22)*D2)
C21
      ANS(36)=ANS(1)*ANS(3)/(ANS(25)*ANS(21)*ANS(22)*D1)
CALC TOTAL FRONT/REAR/INTERCONNECTED RATES = TWICE WHEEL RATES
      DO 1 I=33,36
    1 ANS(I+20)=ANS(I)*2.
      IF(K.LT.0)RETURN
      WRITE(6,100)(ANS(I),I=33,36)
  100 FORMAT( 29H CORRES WHEEL RATES ARE * C11,F10.4,6H * C22,F10.4,6H *
     T C12,F10.4,6H * C21,F10.4,5H KN/M)
      RETURN
      END


      SUBROUTINE FREQ(DL,ANS)
CALC FREQUENCIES FROM SPRINGRATES
      DIMENSION DL(1),ANS(1)
CALC ANS      37    38    39    40    41    42      43      44      45      46
CORRES TO    PFB   PFP   CC1   CC2    PF    BF   PFB**2  PFP**2   PF**2   BF**2
C
CALC ANS      53    54    55    56
CORRES TO   2C11  2C22  2C12  2C21
C
      ANS(43)=(ANS(53)+ANS(54)+ANS(55)+ANS(56))*1E3/DL(45)
      ANS(44)=(DL(42)*DL(42)*ANS(53)+DL(43)*DL(43)*ANS(54)+DL(42)*DL(43)
     T*(ANS(55)+ANS(56)))*1E3/(DL(45)*DL(46)*DL(46))
      ANS(39)=DL(41)*(DL(42)*(ANS(53)+ANS(56))-DL(43)*(ANS(54)+ANS(55)))
     T*1E3/DL(45)
      ANS(40)=(DL(42)*(ANS(53)+ANS(55))-DL(43)*(ANS(54)+ANS(56)))*1E3/(D
     TL(45)*DL(46)*DL(46))
      FT=0.5*(ANS(43)+ANS(44))
      ST=0.5*SQRT((ANS(44)-ANS(43))*(ANS(44)-ANS(43))+4.*ANS(39)*ANS(40)
     T)
      ANS(45)=(FT-ST)
      ANS(46)=(FT+ST)
      DO 2 I=43,46
      IF(ANS(I).LT.0.0)GO TO 1
    2 CONTINUE
      ANS(37)=SQRT(ANS(43))/(2.*3.1415926536)
      ANS(38)=SQRT(ANS(44))/(2.*3.1415926536)
      ANS(41)=SQRT(ANS(45))/(2.*3.1415926536)
      ANS(42)=SQRT(ANS(46))/(2.*3.1415926536)
      WRITE(6,100)ANS(41),ANS(42),(ANS(I),I=37,40)
  100 FORMAT( 25H CORRES FREQS ARE * PITCH,F10.4,9H * BOUNCE,F10.4,3H HZ
     T,5X,6H PFB =,F10.4,6H PFP =,F10.4,3H HZ,5X,11H C COEFFS =,2F10.4)
      RETURN
    1 CONTINUE
      WRITE(6,101)ANS(39),ANS(40),(ANS(I),I=43,46)
  101 FORMAT( 18H FREQS * C COEFFS ,2F10.4,10H * PFB**2 ,F10.4,10H * PFP
     T**2 ,F10.4,9H * PF**2 ,F10.4,9H * BF**2 ,F10.4,8H SEC**-2)
      RETURN
      END


      SUBROUTINE ZERO(I,J,ANS)
CALLED TO ZERO ANS I TO J
      DIMENSION ANS(1)
      DO 1 K=I,J
      ANS(K)=0.0
    1 CONTINUE
      RETURN
      END
```

```
      SUBROUTINE STDEFL(K,DL,ANS)
CALL STDEFL TO CALCULATE STATIC DEFLECTIONS
      DIMENSION DL(1),ANS(1)
      IF(K.GE.0)WRITE(6,100)
  100 FORMAT( /,38H SUBROUTINE * STDEFL * HAS BEEN CALLED)
      CALL ZERO(15,32,ANS)
CALC THT1 AND THT2 AT ZERO LOAD
      ANS(47)=-DL(45)*9.80665*DL(43)*DL(25)*COS(DL(33))/(2000*ANS(1))
      ANS(48)=DL(45)*9.80665*DL(42)*DL(26)*COS(DL(34))/(2000*ANS(3))
CALC F01 AND F02
      ANS(49)=DL(25)*(SIN(DL(33)+ANS(47))-SIN(DL(33)))
      ANS(50)=-DL(26)*(SIN(DL(34)+ANS(48))-SIN(DL(34)))
      CALL DBUMP(0.0,DL,ANS)
CALC G1 AND G2
      ANS(51)=ANS(31)
      ANS(52)=ANS(32)
      IF(K.LT.0)RETURN
      WRITE(6,101)(ANS(I),I=47,50)
  101 FORMAT(16H INITIAL THT1 = ,F10.4,18H * INITIAL THT2 = ,F10.4,
     T/,29H STATIC DEFLECTIONS * FRONT =,F10.4,9H * REAR =,F10.4,2H M)
      RETURN
      END
```

```
      SUBROUTINE STRESS(K,D,A)
CALL STRESS TO CALC STRESS IN TORBARS
CALC WINDUP ANGLES PHI1 AND PHI2
      DIMENSION A(1),D(1)
      A(59)=A(17)-A(19)-A(47)
      A(60)=A(18)-A(20)-A(48)
CALC STRESS S1, S2 AND S3
      A(61)=A(2)*A(59)
      A(62)=A(4)*A(60)
      A(63)=A(6)*A(20)
      WRITE(6,100) (A(I),I=59,63)
  100 FORMAT( ' CORRES ANGLES * PHI1 = ',F10.4,' * PHI2 = ',
     TF10.4,' RAD AND STRESS * S1 = ',F10.4,' * S2 = ',F10.4,
     T' * S3 = ',F10.4,' MPA')
      RETURN
      END
```

APPENDIX 3.2

WISHBONE ANALYSIS PROGRAM, WISHANAL

Purpose. To calculate suspension geometry for a standard four-bar-linkage type wishbone suspension, under static, bounce and rebound conditions.

Input. Seven sets of x,y coordinates that describe the suspension are inputted. The y-axis is the vertical centreline and the x-axis is the groundline. The limits of bounce and rebound are also inputted.

Output. The lengths of the upper and lower suspension arms are printed.

The lengths of various arms and linkages L12, L13, L24, L34, L35, L36 and L37, where for example L12 means the length from the point $(x_1 y_1)$ to the point (x2,y2), are printed.

Various angles of linkage to the x-axis are printed, i.e. AX12, AX13, AX24, AX34, AX35, AX36 and AX37, for the static condition.

Angles A435, A436 and A437 are also outputted, where for example A435 means the angle at point P3 subtended by the line from P4 and P5 in the triangle made up of points P4, P3 and P5.

The kingpin angle is calculated and printed.

Then for each bounce or rebound condition the following output is provided.

K = The deviation in x-coordinate of the upper-outboard balljoint (P3) from its static condition. Bounce is positive, rebound negative.

Y7 = The y-coordinate of the lowest point of the tyre (i.e. tyre/road contact point).

RY = Instantaneous roll, centre height relative to the chassis of the car (i.e. relative to the axes system which assumes that the wheel moves relative to 0,0).

RY-Y7 = Roll centre height relative to the tyre (the y-coordinate of which is Y7).

CAMBER = The camber angle of the tyre, that is, the angle the wheel centreline makes to vertical. Negative camber is the top of the tyre leaning towards the centre of the vehicle. CAMBER assumes no body roll.

SCRUB = The lateral (sideways) displacement of the bottom of the tyre (road interface) from its static position. Scrub is positive outwards.

SARM = The instantaneous swingarm length. That is, the length of the line (perpendicular to the wheel centreline) from the instantaneous swingarm centre (where the top and bottom suspension arms geometrically intersect) to the wheel centreline.

$$\text{RATIO} = \frac{\text{SARM}}{\text{TRACK}} = \frac{\text{swingarm length}}{\text{distance between front wheels}}$$

Terminology

P1 - coordinates (x_1, y_1) is the top pivot affixed to the chassis.

P2 - coordinates (x_2, y_2) is the bottom pivot affixed to the chassis.

P1 and P2 can be actual pivot points or the intersection of the pivot axes with the vertical plane through the front axle centreline.

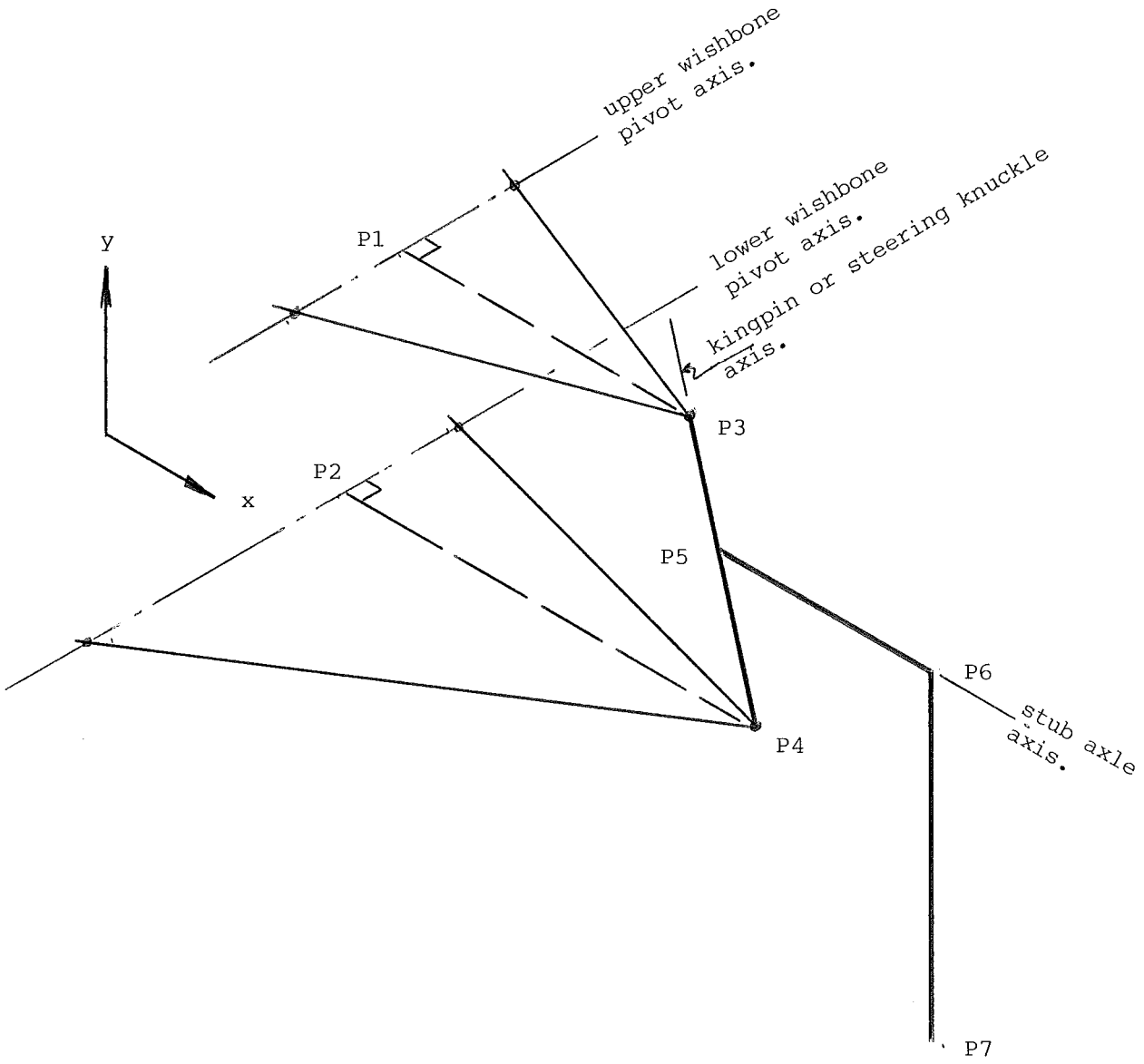
P3 - coordinates (x_3, y_3) is the top pivot or balljoint of the kingpin axis.

P4 - coordinates (x_4, y_4) is the bottom pivot or balljoint of the kingpin axis.

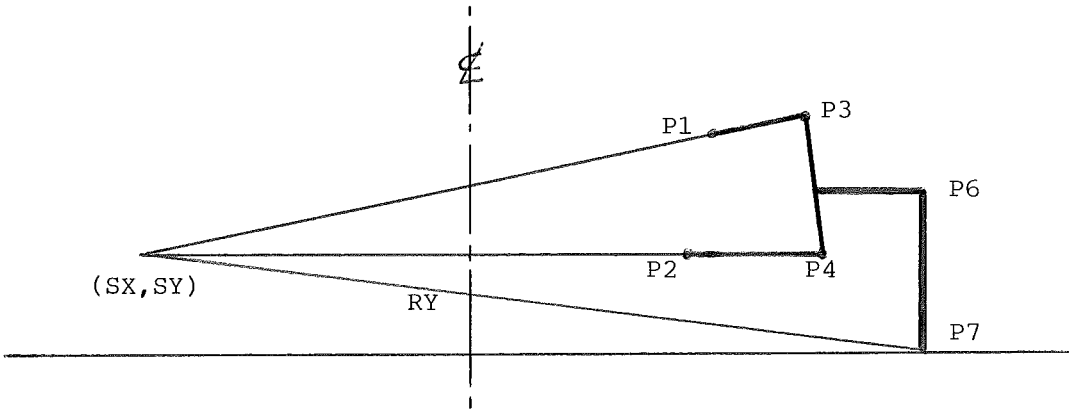
P5 - coordinates (x_5, y_5) is the intersection of the stub axle with the kingpin axis. (Note x_5, y_5 need not be defined for the program to run).

P6 - coordinates (x_6, y_6) is the intersection of the stub axle with the wheel centreline.

P7 - coordinate (x_7, y_7) is the intersection of the wheel centreline with the groundline. i.e. the theoretical centre of the tyre contact patch. In static conditions $y_7 = 0$ and the front track $= 2x_7$.

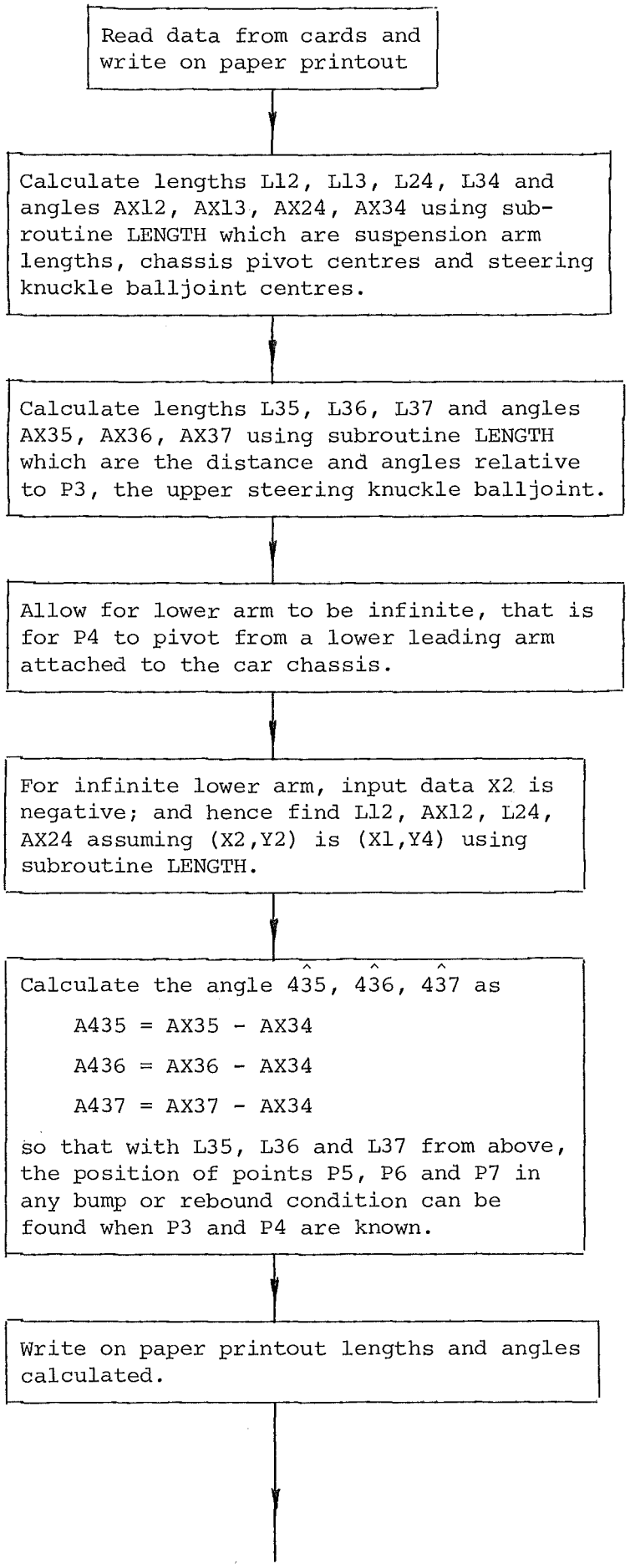


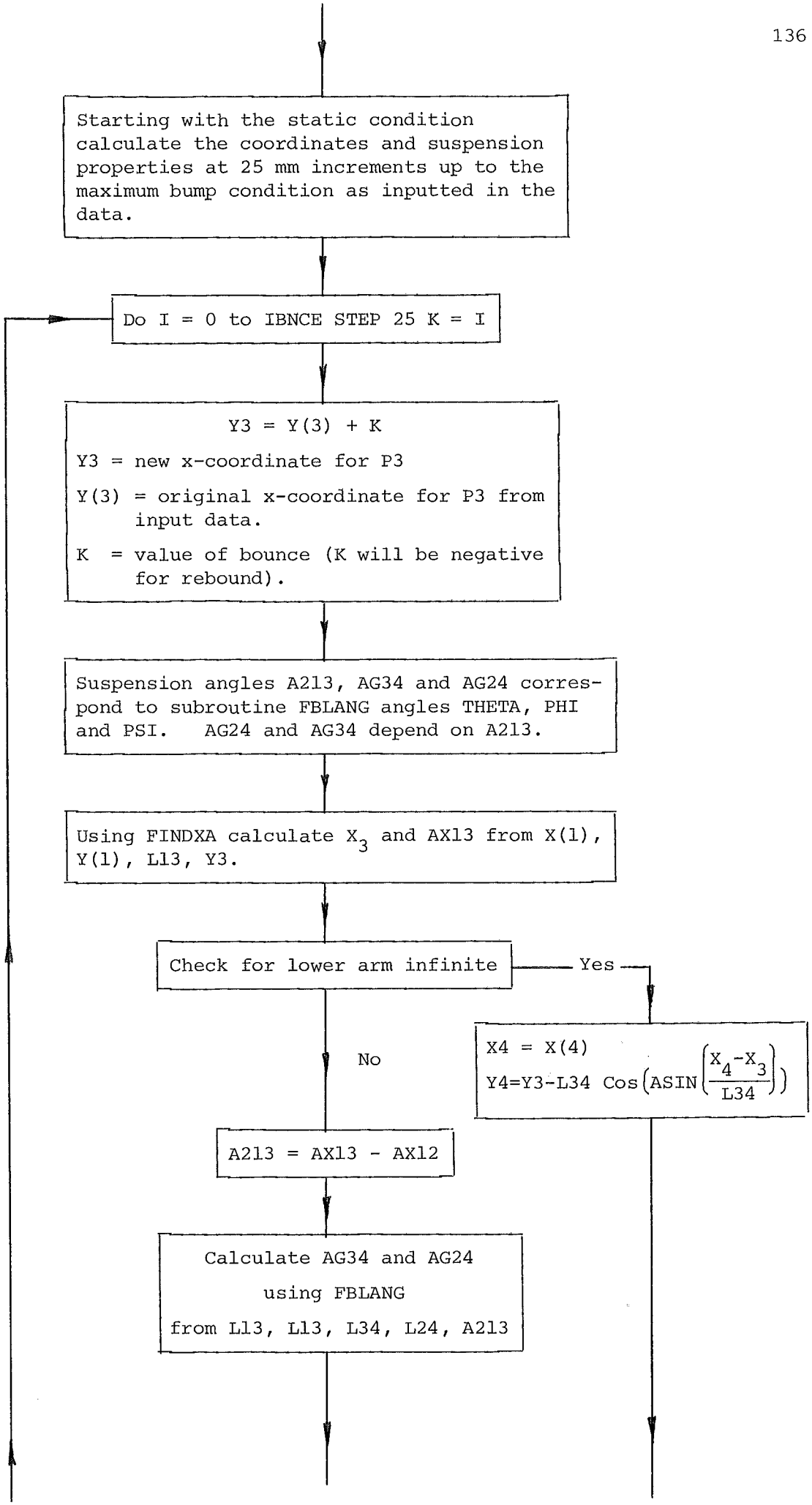
Wishbone Suspension Layout.

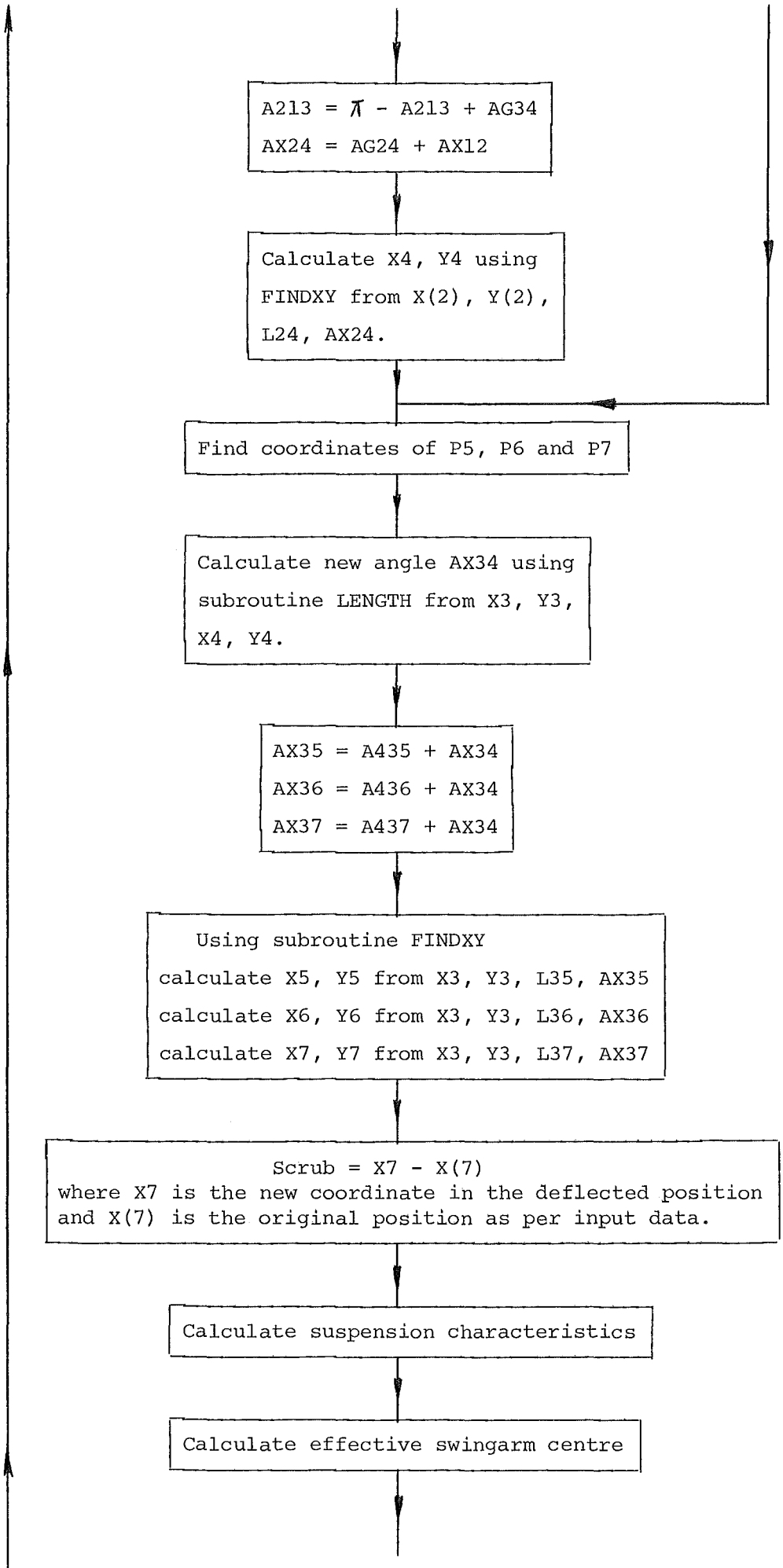


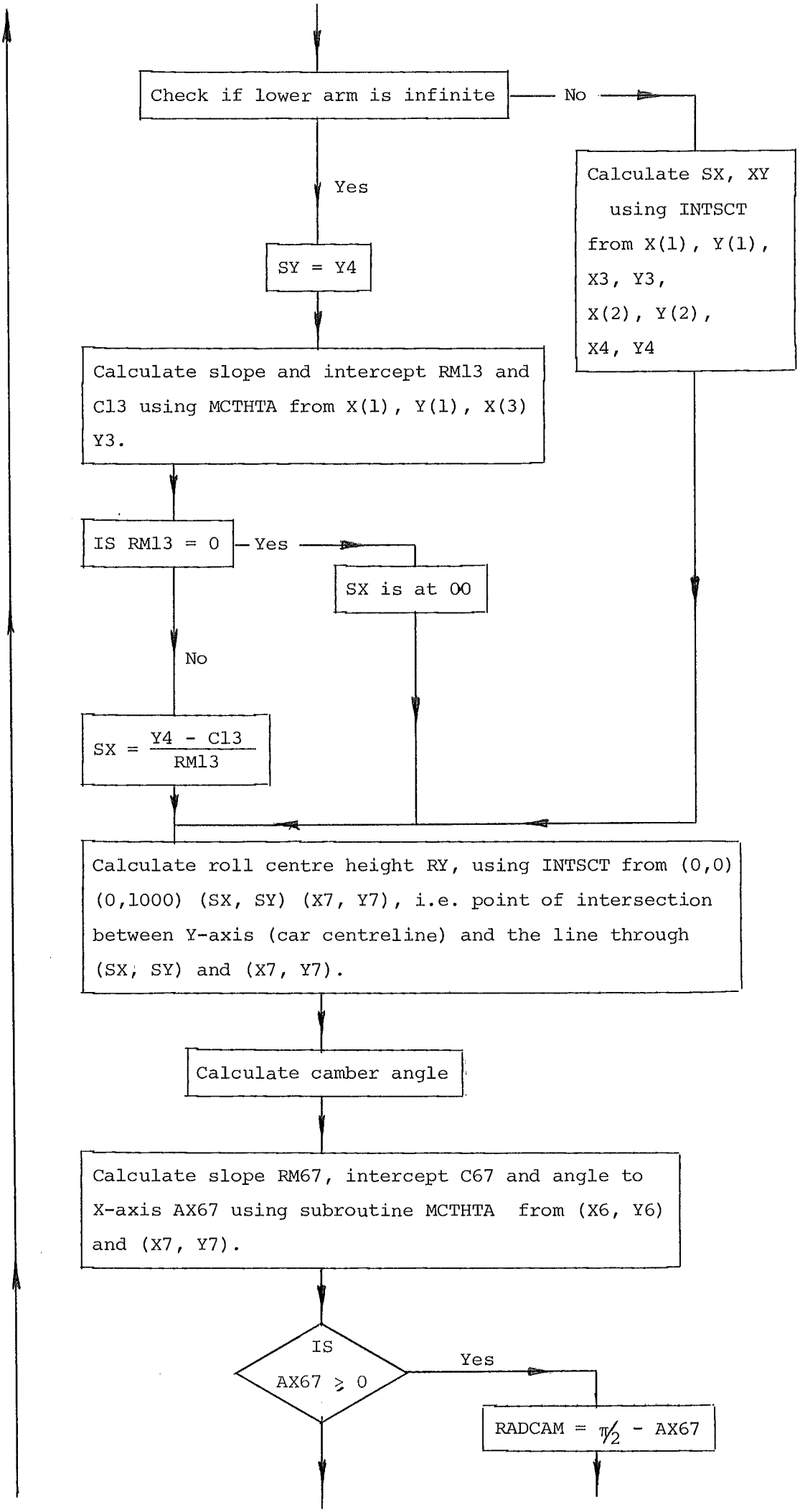
Cross-Section in xy-plane showing swingarm centre and roll centre.

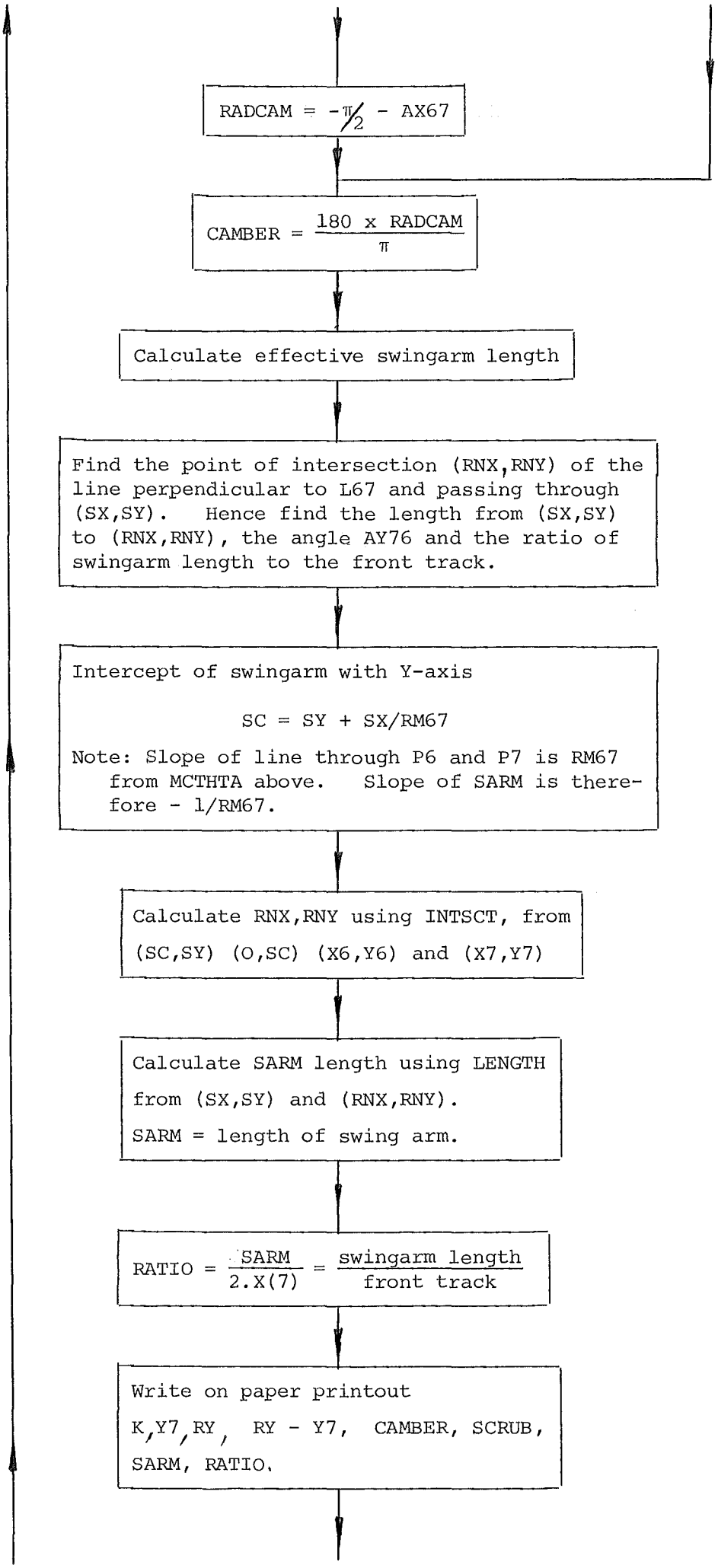
FLOWCHART

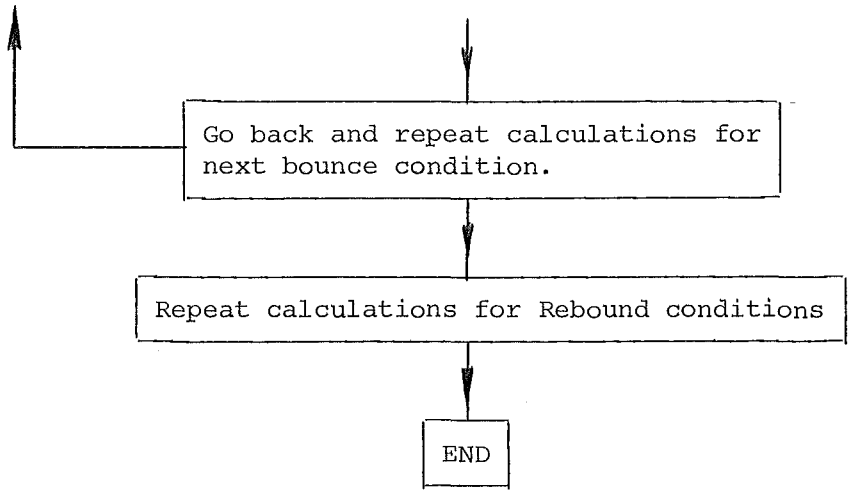

Read data from cards and write on paper printout
Calculate lengths L12, L13, L24, L34 and angles AX12, AX13, AX24, AX34 using sub-routine LENGTH which are suspension arm lengths, chassis pivot centres and steering knuckle balljoint centres.
Calculate lengths L35, L36, L37 and angles AX35, AX36, AX37 using subroutine LENGTH which are the distance and angles relative to P3, the upper steering knuckle balljoint.
Allow for lower arm to be infinite, that is for P4 to pivot from a lower leading arm attached to the car chassis.
For infinite lower arm, input data X2 is negative; and hence find L12, AX12, L24, AX24 assuming (X2,Y2) is (X1,Y4) using subroutine LENGTH.
Calculate the angle 435, 436, 437 as
A435 = AX35 - AX34
A436 = AX36 - AX34
A437 = AX37 - AX34
so that with L35, L36 and L37 from above, the position of points P5, P6 and P7 in any bump or rebound condition can be found when P3 and P4 are known.
Write on paper printout lengths and angles calculated.


Starting with the static condition calculate the coordinates and suspension properties at 25 mm increments up to the maximum bump condition as inputted in the data.
Do I = 0 to IBNCE STEP 25 K = I
Y3 = Y(3) + K
Y3 = new x-coordinate for P3
Y(3) = original x-coordinate for P3 from input data.
K = value of bounce (K will be negative for rebound).
Suspension angles A213, AG34 and AG24 correspond to subroutine FBLANG angles THETA, PHI and PSI. AG24 and AG34 depend on A213.
Using FINDXA calculate X3 and AX13 from X(1), Y(1), L13, Y3.
Check for lower arm infinite
Yes
No
X4 = X(4)
Y4=Y3-L34 Cos(ASIN((X4-X3)/L34))
A213 = AX13 - AX12
Calculate AG34 and AG24
using FBLANG
from L13, L13, L34, L24, A213


A213 = π - A213 + AG34
AX24 = AG24 + AX12
Calculate X4, Y4 using FINDXY from X(2), Y(2), L24, AX24.
Find coordinates of P5, P6 and P7
Calculate new angle AX34 using subroutine LENGTH from X3, Y3, X4, Y4.
AX35 = A435 + AX34
AX36 = A436 + AX34
AX37 = A437 + AX34
Using subroutine FINDXY
calculate X5, Y5 from X3, Y3, L35, AX35
calculate X6, Y6 from X3, Y3, L36, AX36
calculate X7, Y7 from X3, Y3, L37, AX37
Scrub = X7 - X(7)
where X7 is the new coordinate in the deflected position
and X(7) is the original position as per input data.
Calculate suspension characteristics
Calculate effective swingarm centre


Check if lower arm is infinite
No
Yes
Calculate SX, XY using INTSCT from X(1), Y(1), X3, Y3, X(2), Y(2), X4, Y4
SY = Y4
Calculate slope and intercept RM13 and C13 using MCTHTA from X(1), Y(1), X(3) Y3.
IS RM13 = 0
Yes
SX is at 00
No
SX = Y4 - C13 / RM13
Calculate roll centre height RY, using INTSCT from (0,0) (0,1000) (SX, SY) (X7, Y7), i.e. point of intersection between Y-axis (car centreline) and the line through (SX, SY) and (X7, Y7).
Calculate camber angle
Calculate slope RM67, intercept C67 and angle to X-axis AX67 using subroutine MCTHTA from (X6, Y6) and (X7, Y7).
IS AX67 ≥ 0
Yes
RADCAM = π/2 - AX67


RADCAM = $-\pi/2$ - AX67
CAMBER = $\frac{180 \times RADCAM}{\pi}$
Calculate effective swingarm length
Find the point of intersection (RNX,RNY) of the line perpendicular to L67 and passing through (SX,SY). Hence find the length from (SX,SY) to (RNX,RNY), the angle AY76 and the ratio of swingarm length to the front track.
Intercept of swingarm with Y-axis
SC = SY + SX/RM67
Note: Slope of line through P6 and P7 is RM67 from MCTHTA above. Slope of SARM is therefore - 1/RM67.
Calculate RNX,RNY using INTSCT, from (SC,SY) (O,SC) (X6,Y6) and (X7,Y7)
Calculate SARM length using LENGTH from (SX,SY) and (RNX,RNY).
SARM = length of swing arm.
RATIO = $\frac{SARM}{2.X(7)}$ = $\frac{\text{swingarm length}}{\text{front track}}$
Write on paper printout
K, Y7, RY, RY - Y7, CAMBER, SCRUB, SARM, RATIO.

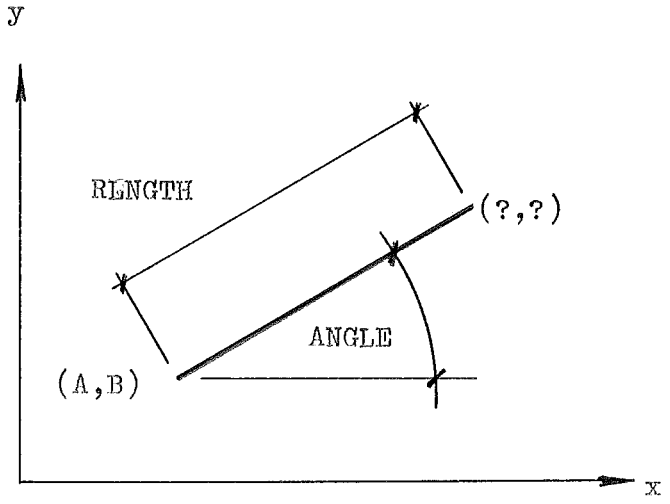

Go back and repeat calculations for next bounce condition.
Repeat calculations for Rebound conditions
END

SUBROUTINE FINDXY

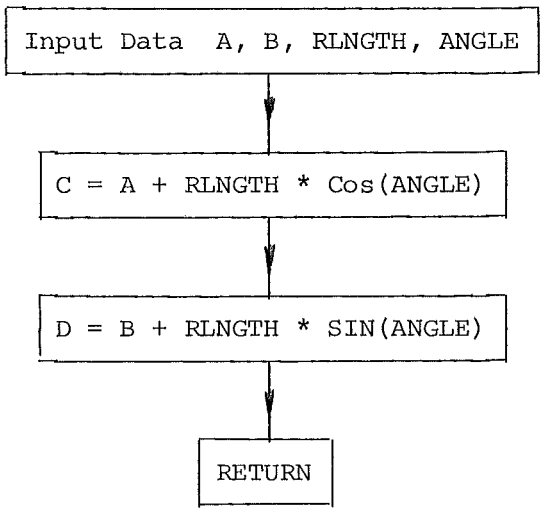
Purpose. To find the coordinates of a point given the length and angle of a line from another known point.

Input. Coordinates of known point (A,B) length RLNGTH and angle ANGLE.

Output. Coordinates (C,D)



FLOWCHART

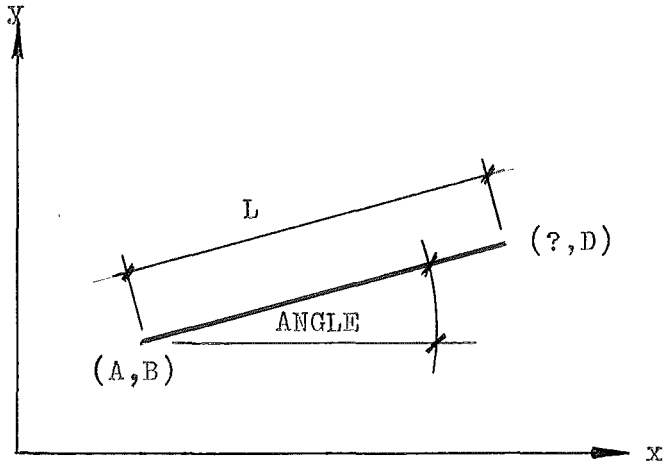


SUBROUTINE FINDXA

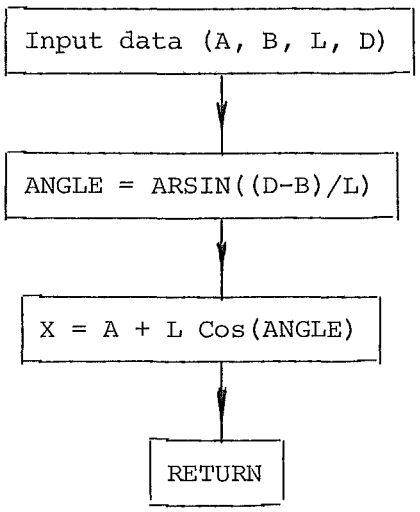
Purpose. To find the x-coordinate of a point, given the y-coordinate, and the length of the line from another known point. Also to find the angle to the x-axis of the said line.

Input. The y-coordinate D, the length of the line L and the coordinates of the known point (A,B).

Output. The x-coordinate, x and angle THETA.



FLOWCHART



SUBROUTINE FBLANG

Purpose. To find the output angles for a Four-Bar-Linkage.

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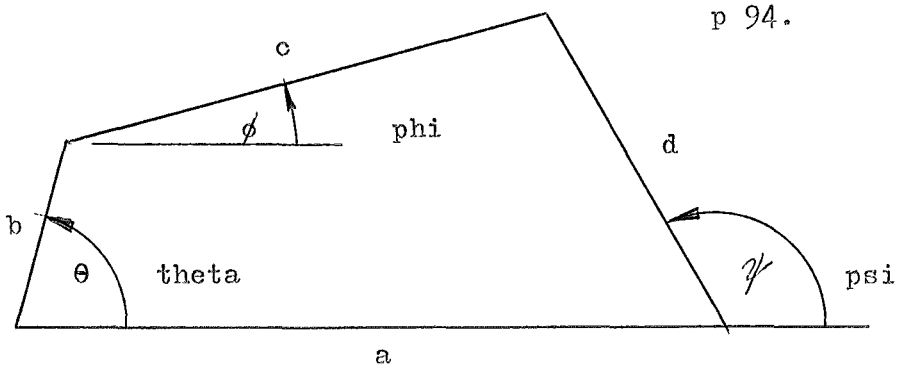


$$\phi = \pm \cos^{-1}\left\{\frac{(a^2+b^2 - 2ab\cos\theta) + c^2 - d^2}{2c\sqrt{a^2 + b^2 - 2ab\cos\theta}}\right\} - \tan^{-1}\left\{\frac{b\sin\theta}{a-b\cos\theta}\right\}$$

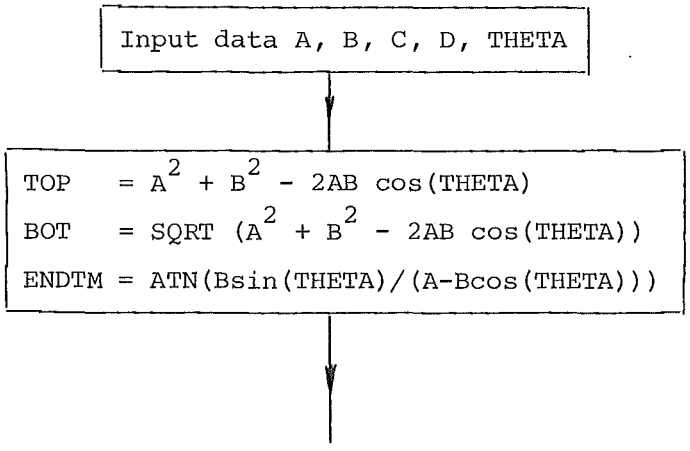
$$\psi = 180^{o} \pm \cos^{-1}\left\{\frac{(a^2+b^2 - 2ab\cos\theta) + d^2 - c^2}{2d\sqrt{a^2+b^2 - 2ab\cos\theta}}\right\} - \tan^{-1}\left\{\frac{b\sin\theta}{a-b\cos}\right\}$$

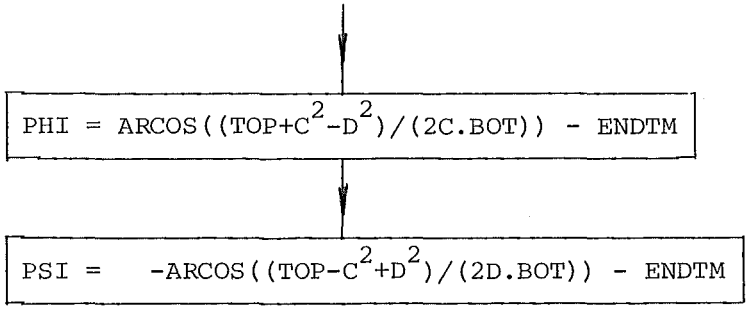
FBLANG means Four Bar Linkage Angles.

Input. Four lengths A, B, C, D and input angle θ (THETA).

Output. Angles ϕ(PHI) and ψ(PSI).

FLOWCHART




PHI = ARCOS((TOP+C2-D2)/(2C.BOT)) - ENDTM
PSI = -ARCOS((TOP-C2+D2)/(2D.BOT)) - ENDTM

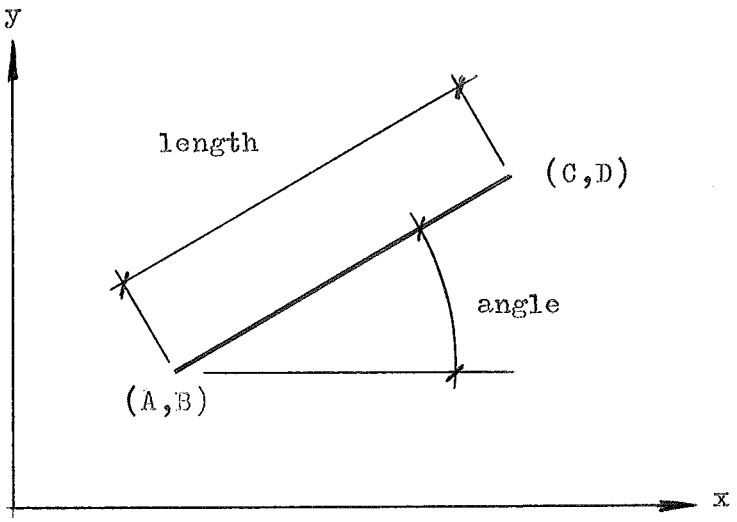
SUBROUTINE LENGTH

Purpose. To find the length of the line between two points and the angle that line makes with the x-axis.

Input. Two pairs of coordinates (A,B) and (C,D).

Output. The length and angle.

Discussion.



$$\text{angle} = \text{ATAN}\left(\frac{D-B}{C-A}\right)$$

must check for $0 < \text{angle} < 360$.

1. If $(D-B) > 0 + (C-A) > 0$ then $0 < \text{angle} < 90$.

2. If $(D-B) > 0 + (C-A) < 0$ then $270 < \text{angle} < 360$, or $-90 < \text{angle} < 0$

3. If $(D-B) < 0 + (C-A) > 0$ then $90 < \text{angle} < 180$.

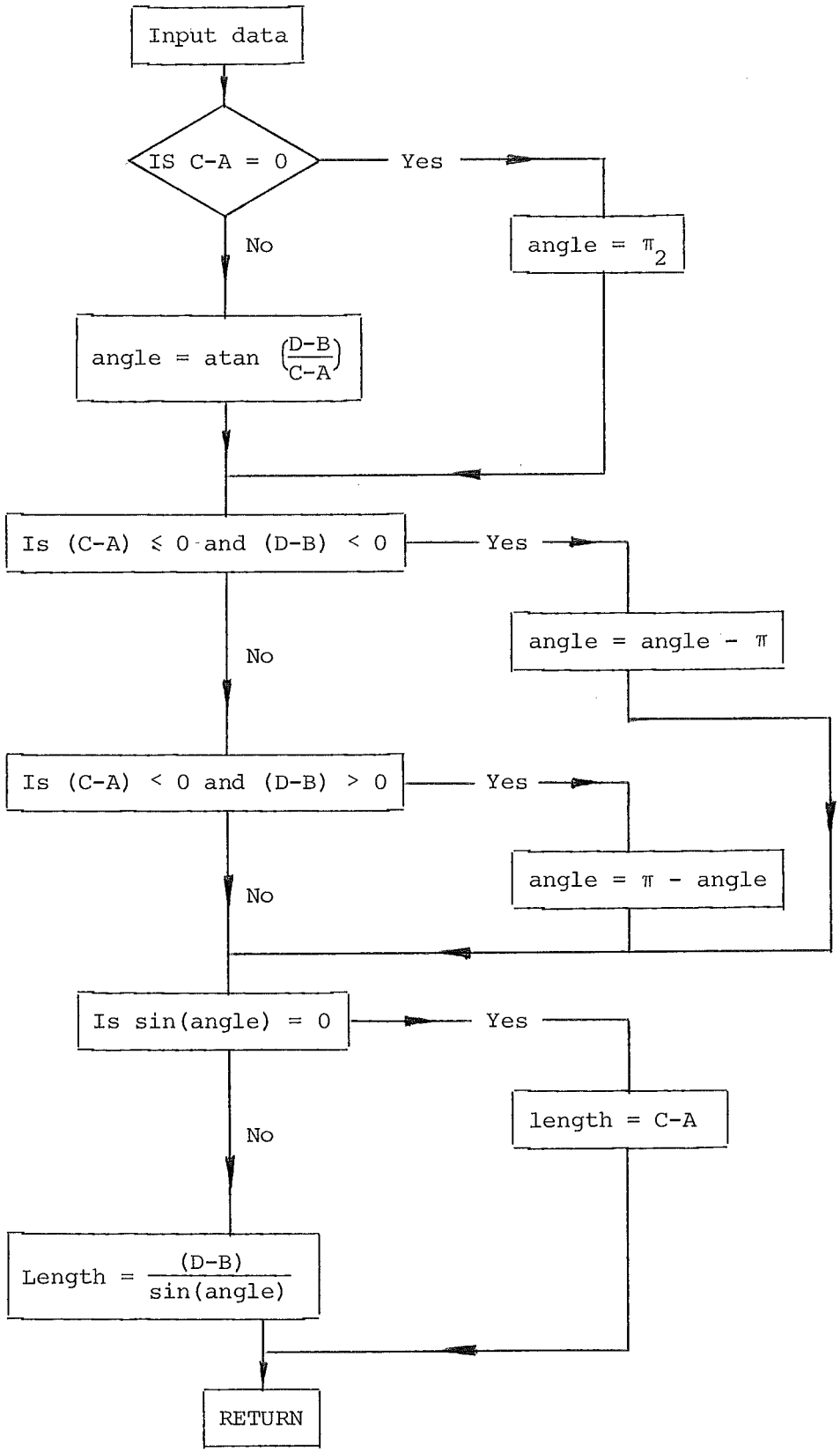
4. If $(D-B) < 0 + (C-A) < 0$ then $180 < \text{angle} < 270$.

Given the angle, then

$$\text{Length} = \frac{(D-B)}{\sin(\text{angle})}$$

except that if $\sin(\text{angle}) = 0$ then length = (C-A).

FLOWCHART


Input data
IS C-A = 0
Yes
No
angle = π/2
angle = atan (D-B)/(C-A)
Is (C-A) ≤ 0 and (D-B) < 0
Yes
No
angle = angle - π
Is (C-A) < 0 and (D-B) > 0
Yes
No
angle = π - angle
Is sin(angle) = 0
Yes
No
length = C-A
Length = (D-B)/sin(angle)
RETURN

SUBROUTINE MCTHTA

Purpose. To find the slope and intercept of a line y = Mx + C defined by two points. From the slope find also the angle to the x-axis.

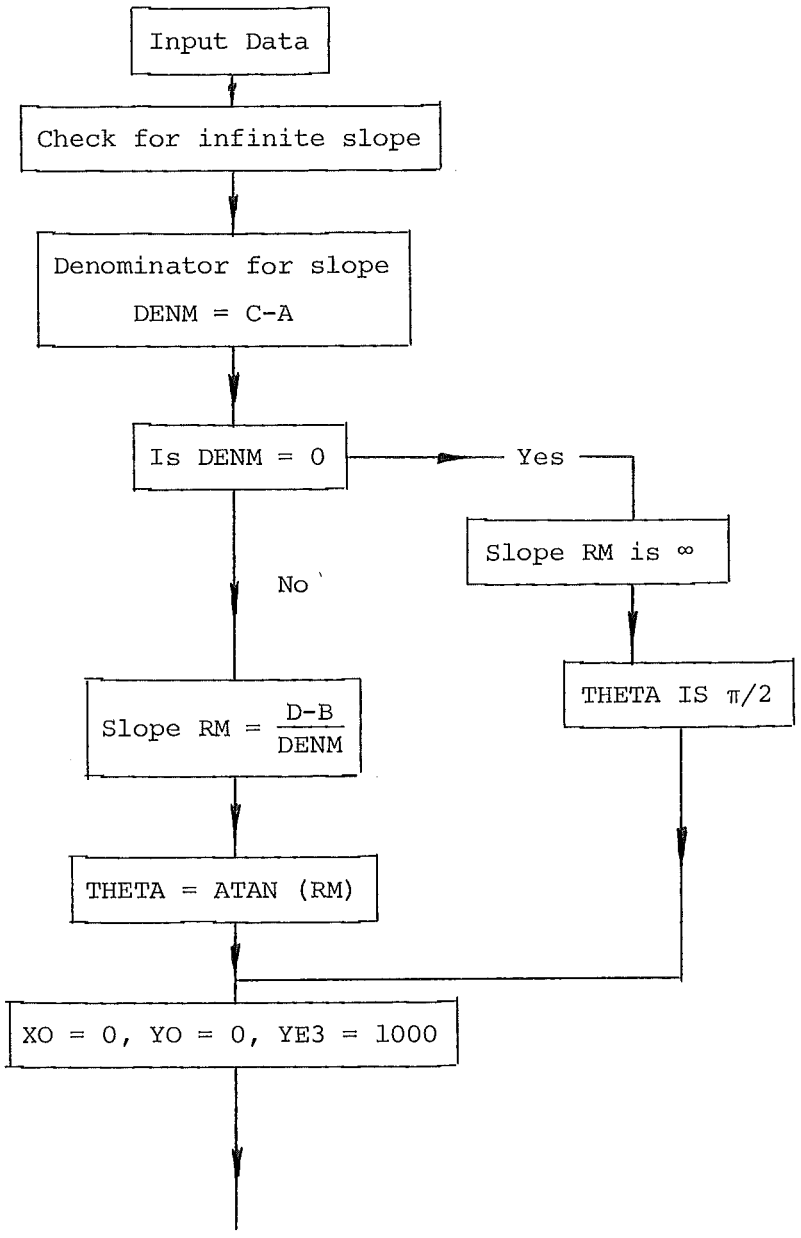
Input. Two pairs of coordinates (A,B) (C,D).

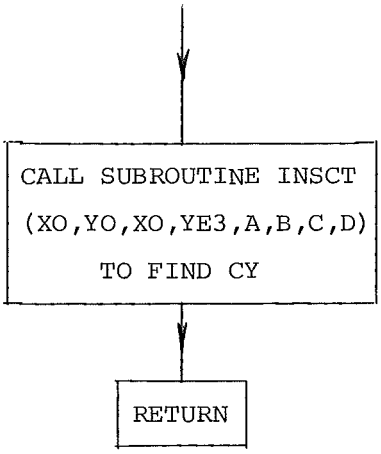
Output. Slope RM, intercept DY and angle THETA.

Discussion.

From coordinates (A,B) and (C,D) find the line y = Mx + C. The slope RM = $\frac{D-B}{C-A}$, so that the angle THETA = ATAN (RM) and the y-intercept is found using subroutine INTSCT.

FLOWCHART




CALL SUBROUTINE INSCT
(XO,YO,XO,YE3,A,B,C,D)
TO FIND CY
RETURN

SUBROUTINE INTSCT

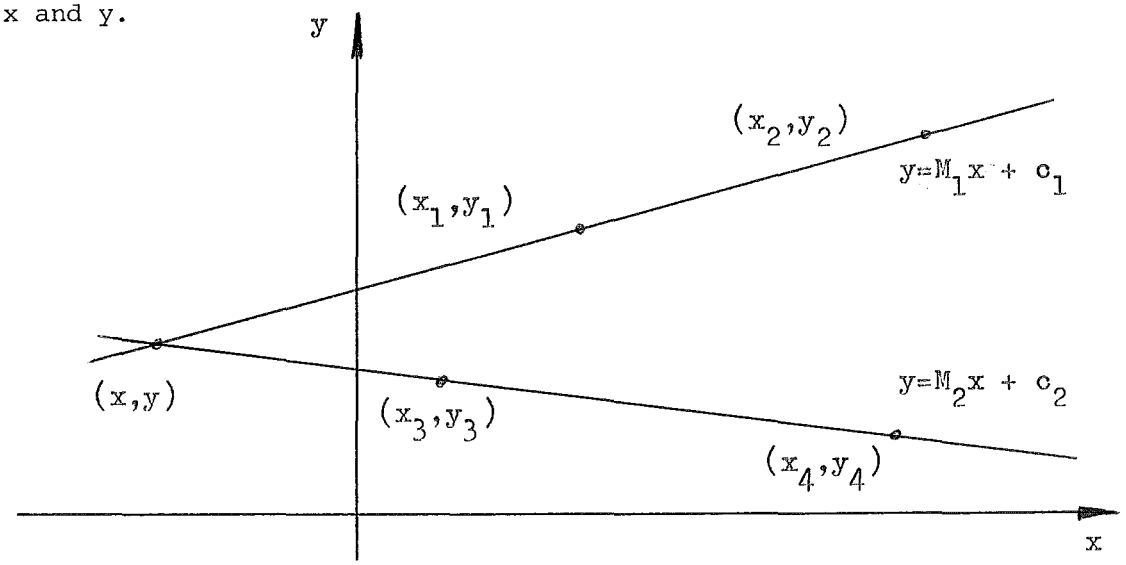
Purpose. To find the point of intersection (x,y) of two lines. Each line is defined by two pair of points.

Input. Four pairs of coordinates (x_1y_1) (x_2y_2) and (x_3y_3) (x_4y_4).

Output. One point of intersection (x,y).

Discussion.

Define the two lines in the form of y = mx + c and solve for x and y.



$$M_1 = \frac{y_2 - y_1}{x_2 - x_1} \qquad M_2 = \frac{y_4 - y_3}{x_4 - x_3}$$

$$C_1 = \frac{x_2y_1 - x_1y_2}{x_2 - x_1} \qquad C_2 = \frac{x_4y_3 - x_3y_4}{x_4 - x_3}$$

$$x = \frac{C_2-C_1}{M_1-M_2} = \frac{(x_2-x_1)(x_4y_3-x_3y_4) - (x_4-x_3)(x_2y_1-x_1y_2)}{(x_4-x_3)(y_2-y_1) - (x_2-x_1)(y_4-y_3)}$$

or as in program

$$X = \frac{(x_3-x_4)(y_1x_2-x_1y_2) - (x_1-x_2)(y_3x_4-x_3y_4)}{(y_1-y_2)(x_3-x_4) - (y_3-y_4)(x_1-x_2)}$$

From $\dfrac{y_2-y}{x_2-x} = \dfrac{y_1-y}{x_1-x}$

$$\| \quad y = \frac{y_1(x-x_2) - y_2(x-x_1)}{(x_1-x_2)}$$

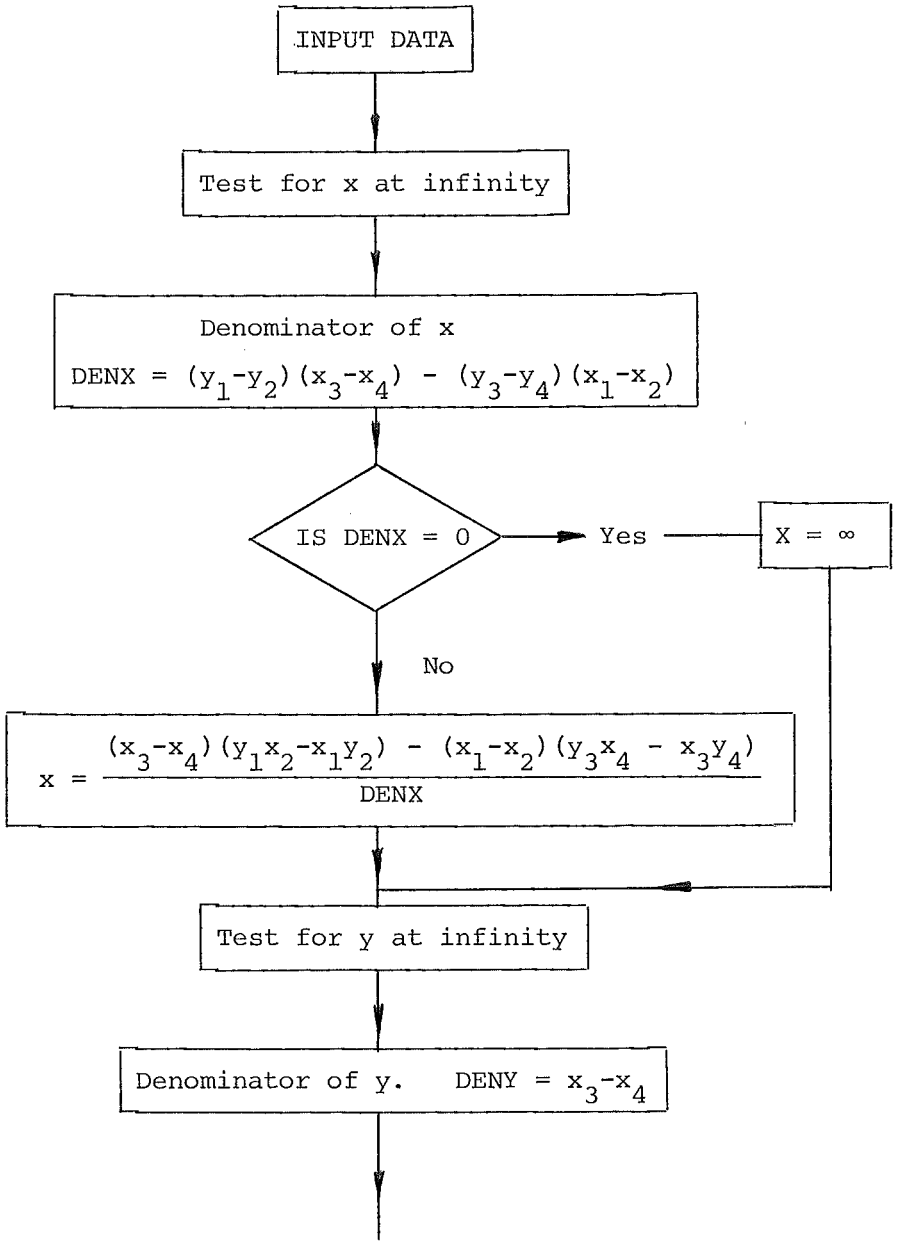
similarly

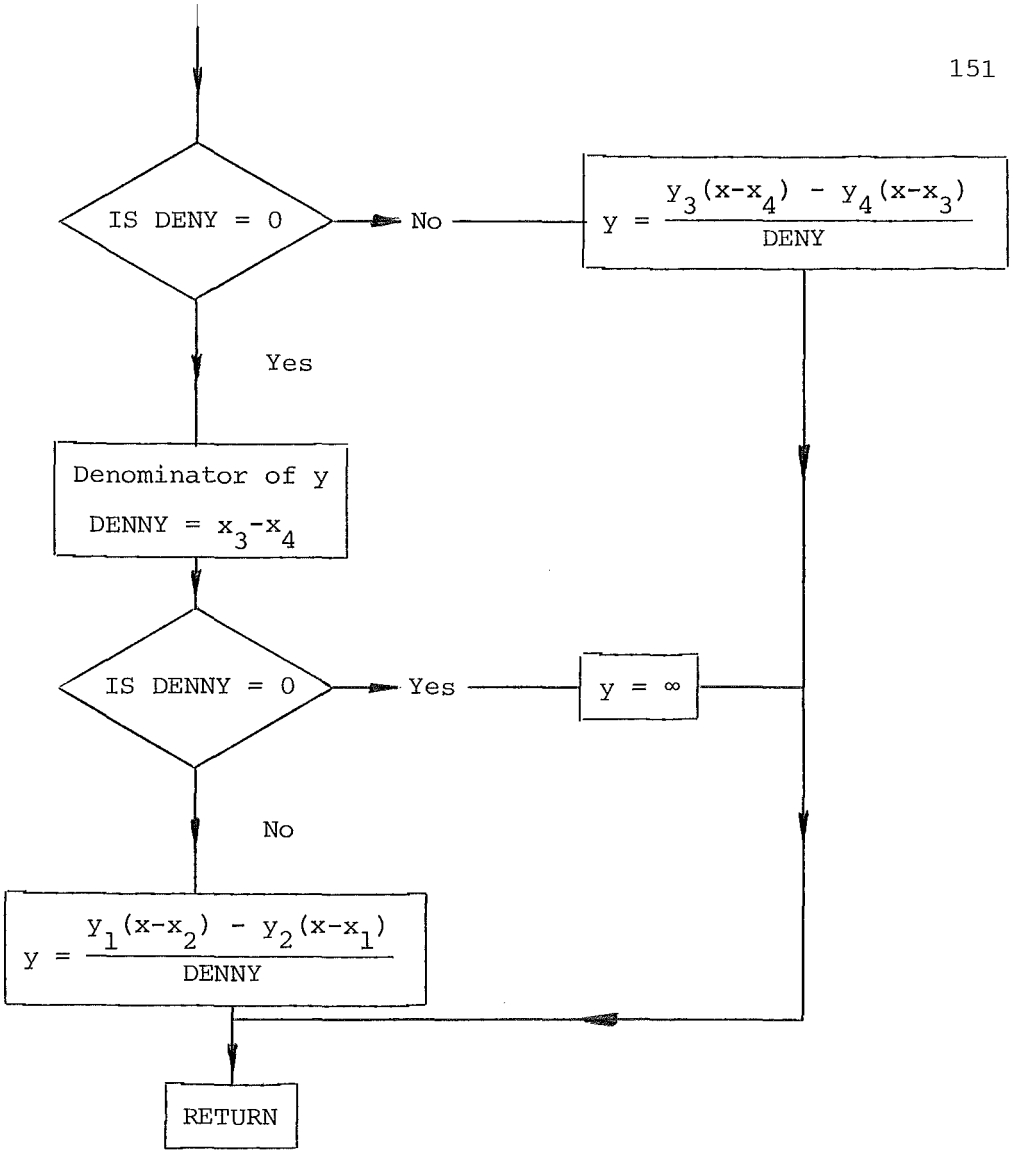
$$\| \quad y = \frac{y_3(x-x_4) - y_4(x-x_3)}{(x_3-x_4)}$$

If $(y_1-y_2)(x_3-x_4) - (y_3-y_4)(x_1-x_2) = 0$ then x is at infinity.

If $(x_3-x_4) = 0$ then check if $(x_1-x_2) = 0$ if both then y is at infinity.

FLOWCHART




IS DENY = 0
No
$y = \frac{y_3(x-x_4) - y_4(x-x_3)}{DENY}$
Yes
Denominator of y
DENNY = x_3-x_4
IS DENNY = 0
Yes
y = ∞
No
$y = \frac{y_1(x-x_2) - y_2(x-x_1)}{DENNY}$
RETURN

```
C
C PRELIMINARY ANALYSIS OF WISHBONE SUSPENSION GEOMETRY
C

      DIMENSION X(7),Y(7)
C  DATA READ IN ON ONE CARD SEPERATED BY COMMAS
C  PLUS ANOTHER CARD FOR BOUNCE,REBOUND
C  MUST HAVE CARD WITH 14 COMMAS AFTER DATA AND BEFORE END JOB
C   FOR LOWER ARM INFINITE * SPECIFY X2 AS NEGATIVE
C
    1 CONTINUE
      READ(5,2)(X(I),Y(I),I=1,7)

      002:0012:1  IS THE LOCATION FOR EXCEPTIONAL ACTION ON THE I/O STATE
      IF(X(1).EQ.0.0) GO TO 99
      READ(5,2)IBNCE,IREBND
      002:001E:1  IS THE LOCATION FOR EXCEPTIONAL ACTION ON THE I/O STATE
      WRITE (6,101)

      WRITE (6,103)
      WRITE (6,102)(I,X(I),Y(I),I=1,7)
      WRITE(6,100)
      WRITE(6,107)IBNCE,IREBND
C  L13 IS THE UPPER WISHBONE LENGTH, L24 IS THE LOWEE WISHBONE LENGTH
C   A , B , C , D  ARE LENGTHS IN A FOUR BAR LINKAGE CORRES TO LENGTHS
C  L12,L13,L34,L24 FOR THE SUSPENSION
C
C  NOTE ALL ANGLES IN RADIANS; EXCEPT CAMBER AND AKPIN
C
C  L12,L13,L34,L24 REMAIN CONSTANT THROUGHOUT
C   ANGLE  AX12  REMAINS CONSTANT THROUGHOUT
      CALL LENGTH(X(1),Y(1),X(2),Y(2),L12,AX12)
      CALL LENGTH(X(1),Y(1),X(3),Y(3),L13,AX13)
      CALL LENGTH(X(2),Y(2),X(4),Y(4),L24,AX24)
      CALL LENGTH(X(3),Y(3),X(4),Y(4),L34,AX34)
      WRITE(6,106)L13,L24
C  TO CALC A435 ETC AND L35 ETC
      CALL LENGTH(X(3),Y(3),X(5),Y(5),L35,AX35)
      CALL LENGTH(X(3),Y(3),X(6),Y(6),L36,AX36)
      CALL LENGTH(X(3),Y(3),X(7),Y(7),L37,AX37)
      WRITE(6,300)
      WRITE(6,210)L12,L13,L24,L34,L35,L36,L37
      WRITE(6,301)
      WRITE(6,210)AX12,AX13,AX24,AX34,AX35,AX36,AX37
C  ALLOW FOR INFINITE LOWER ARM
      IF (X(2))7,8,8
    7 CONTINUE
      CALL LENGTH(X(1),Y(1),X(4),Y(4),L12,AX12)
      CALL LENGTH(X(1),Y(4),X(4),Y(4),L24,AX24)
      WRITE(6,307)L12,L24,AX12,AX24
    8 CONTINUE
      A435=AX35-AX34
      A436=AX36-AX34
      A437=AX37-AX34
      WRITE(6,303)
      WRITE(6,211)A435,A436,A437
C CALCULATE AKPIN = KINGPIN ANGLE
      CALL LENGTH(X(6),Y(6),X(7),Y(7),L67,AX67)
      AKPIN=180.0*(AX34-AX67)/3.1415926536
      WRITE(6,213)AKPIN
      WRITE(6,212)
C  ABOVE THIS POINT , VALUES ARE CONSTANT
```

```
C
C  CALCULATE NEW SUSPENSION COORDINATES
C
C
C
      WRITE(6,104)
      N=0
      DO 20 I=0,IBNCE,25
      K=1
    5 CONTINUE
      DO 21 J=0,N,25
      IF (N.EQ.0)GO TO 6
      K=-J
    6 CONTINUE
      Y3=Y(3)+K
C   ANGLES A213,AG34,AG24 IN THE SUSPENSION CORRES TO
C   ANGLES THETA, PHI,PSI IN THE FOUR BAR LINKAGE
C
C   ANGLES A134,AG24 DEPEND ON THE ANGLE A213
C   FIND THE NEW X3 VALUE
      CALL FINDXA(X(1),Y(1),L13,Y3,X3,AX13)
C   NOTE: POINTS FOUND ARE IN QUANDRANTS ONE AND FOUR ONLY
C   CHECK FOR INFINITE LOWER ARM
      IF (X(2)9,10,10
    9 CONTINUE
      X4=X(4)
      Y4=Y3-L34*COS(ARSIN((X4-X3)/L34))
      GO TO 11
   10 CONTINUE
C   FIND A213 = THETA
      A213=AX13-AX12
      CALL FBLANG(L12,L13,L34,L24,A213,AG34,AG24)
      A134=3.1415926530-A213+AG34
      AX24=AG24+AX12
C   FIND X4,Y4
      CALL FINDXY(X(2),Y(2),L24,AX24,X4,Y4)
   11 CONTINUE
C   FIND COORDS OF POINTS 5,6, AND 7
C   CALC NEW ANGLE  AX34
      CALL LENGTH(X3,Y3,X4,Y4,CL34,AX34)
      AX35=A435+AX34
      AX36=A436+AX34
      AX37=A437+AX34
      CALL FINDXY(X3,Y3,L35,AX35,X5,Y5)
      CALL FINDXY(X3,Y3,L36,AX36,X6,Y6)
      CALL FINDXY(X3,Y3,L37,AX37,X7,Y7)
      SCRUB=X7-X(7)
```

```
C  CALC SUSP CHARACTERISTICS
C
C
C
C  CALC EFFECTIVE SARM CENTRE = S
      IF (X(2))12,13,13
   12 CONTINUE
      CALL MCTHTA(X(1),Y(1),X3,Y3,RM13,C13,ANG13)
      SY=Y4
      IF(RM13)15,16,15
   16 SX=1.E40
      GO TO 14
   15 CONTINUE
      SX=(Y4-C13)/RM13
      GO TO 14
   13 CONTINUE
      CALL INTSCT(X(1),Y(1),X3,Y3,X(2),Y(2),X4,Y4,SX,SY)
   14 CONTINUE
C  CALC ROLL CENTRE POSITION = R    ROLL CENTRE HEIGHT = RY
      X0=0.0
      Y0=0.0
      YE3=1000.0
      CALL INTSCT(X0,Y0,X0,YE3,SX,SY,X7,Y7,RX,RY)
C  CALC CAMBER ANGLE
C RM67 IS THE SLOPE THRU P6 AND P7
      CALL MCTHTA(X6,Y6,X7,Y7,RM67,C67,AX67)
      IF(AX67.GE.0.0)RADCAM=1.5707963268-AX67
      IF(AX67.LT.0.0)RADCAM=-1.5707963268-AX67
      CAMBER=180.0*RADCAM/3.1415926536
C  CALC EFFECTIVE SARM LENGTH = SARM
C  REQD TO FIND LGTH OF LINE THRU P6 ANDP7 AND PASSING THRU S
      SC=SY+SX/RM67
      CALL INTSCT(SX,SY,X0,SC,X6,Y6,X7,Y7,RNX,RNY)
      CALL LENGTH(SX,SY,RNX,RNY,SARM,NGLARM)
      RATIO=SARM/(2*X(7))
      RYDASH=RY-Y7
      WRITE(6,105)K,Y7,RY,RYDASH,CAMBER,SCRUB,SARM,RATIO
   21 CONTINUE
   20 CONTINUE
      IF(IREBND.EQ.0)GO TO 1
      N=IREBND
      IF (IBNCE.EQ.0.AND.K.EQ.0) K=1
      IF (K) 1,1,5
   99 CONTINUE
  100 FORMAT( //,' TO CALCULATE FOR  BOUNCE   AND  REBOUND')
  101 FORMAT( //,' PRELIMINARY ANALYSIS OF WISHBONE SUSPENSION GEOMETRY
     T ')
  102 FORMAT( 5X,I2,5X,F9.1,5X,F9.1)
  103 FORMAT(013X,' X-COORD',7X,' Y-COORD')
  104 FORMAT( /,5X,'K',6X,'Y7',8X,'RY',7X,'RY-Y7',4X,'CAMBER',4X,'SCRUB'
     T,6X,'SARM',5X,'RATIO')
  105 FORMAT( ' *',I4,7F10.3)
  106 FORMAT( /,' LENGTH OF UPPER ARM IS ', F8.3,/
     T/,' LENGTH OF LOWER ARM IS ' F8.3)
  107 FORMAT( 20X,I5,10X,I5)
  200 FORMAT( /,2F15.4)
  201 FORMAT( /,3F15.4)
  202 FORMAT( /,F15.3)
  203 FORMAT( /,E15.4)
  204 FORMAT( /,6F15.6)
  210 FORMAT( 10X,8F10.3)
  211 FORMAT( 10X,3F10.3)
  212 FORMAT( /,' SCRUB IS POSITIVE OUTWARDS ')
  213 FORMAT( /,' KINGPIN ANGLE  = ', F10.3)
  214 FORMAT( 10X,10F10.3)
  300 FORMAT( /,' LENGTHS',8X,'L12',7X,'L13',7X,'L24',7X,'L34',7X,'L35',
     T7X,'L36',7X,'L37')
  301 FORMAT( /,' ANGLES',8X,'AX12',6X,'AX13',6X,'AX24',6X,'AX34',6X,
     T'AX35',6X,'AX36',6X,'AX37')
  303 FORMAT( /,' ANGLES',9X,'A435',7X,'A436',7X,'A437')
  304 FORMAT( /,' ANGLES',9X,'A213',7X,'AG34',7X,'AG24')
  305 FORMAT( /,' NEW COORDS',3X,'X3',8X,'Y3',8X,'X4',8X,'Y4',8X,'X5',
     T8X,'Y5',8X,'X6',8X,'Y6',8X,'X7',8X,'Y7')
  306 FORMAT( ' ANGLES',7X,'A213',6X,'AX13',6X,'AG24')
  307 FORMAT( /,' LOWER ARM IS OF INFINITE LENGTH ',/,
     T' NEW CALC LENGTHS *  L12 =',F10.3,'  L24 =',F10.3,
     T/,18X,'* AX13 =',F10.3,'  AX24 =',F10.3)
  308 FORMAT( ' SARM CENTRE',2F15.3,' ROLL CENTRE',2F10.3)
 1000 FORMAT ( 7F10.3,/,7F10.3)
 1001 FORMAT( 2I5)
      END
```

```
      SUBROUTINE   INTSCT(X1,Y1,X2,Y2,X3,Y3,X4,Y4,X,Y)
C   (X,Y) IS THE POINT OF INTERSECTION OF TWO LINES
C   TEST FOR  X AT INFINITY
      DENX=((Y1-Y2)*(X3-X4)-(Y3-Y4)*(X1-X2))
      IF (DENX) 2,1,2
    1 X=1E40
      GO TO 5
    2 CONTINUE
      X=((X3-X4)*(Y1*X2-X1*Y2)-(X1-X2)*(Y3*X4-X3*Y4))/
     1DENX
    5 CONTINUE
C   TEST THAT Y IS NOT AT INFINITY
      DENY=(X3-X4)
      IF (DENY.EQ.0.0) GO TO 3
      Y=(Y3*(X-X4)-Y4*(X-X3))/DENY
      GO TO 6
    3 CONTINUE
      DENNY=(X1-X2)
      IF (DENNY.EQ.0.0) GO TO 4
      Y=(Y1*(X-X2)-Y2*(X-X1))/DENNY
      GO TO 6
    4 Y=1E40
    6 CONTINUE
      RETURN
      END


      SUBROUTINE MCTHTA(A,B,C,D,RM,CY,THETA)
C   M, C, THETA ARE SLOPE, INTERCEPT, THETA RESP.
C    A,B,C,D ARE THE COORDS OF THE TWO POINTS
      DENM=(C-A)
      IF (DENM ) 2,1,2
    1 CONTINUE
      RM=1E40
      THETA=1.5707963268
      GO TO 3
    2 RM=(D-B)/DENM
      THETA=ATAN(RM)
    3 CONTINUE
      X0=0.0
      Y0=0.0
      YE3=1000.0
      CALL INTSCT(X0,Y0,X0,YE3,A,B,C,D,CX,CY)
      RETURN
      END


      SUBROUTINE LENGTH(A,B,C,D,RLNGTH,ANGLE)
C   LENGTH FINDS LENGTH, ANGLE GIVEN TWO POINTS
C    A,B,C,D ARE THE COORDS OF THE TWO POINTS
      IF ((C-A).EQ.0.0) GO TO 1
      ANGLE=ATAN((D-B)/(C-A))
      GO TO 2
    1 ANGLE=1.5707963268
    2 CONTINUE
      IF((C-A).LE.0.0.AND.(D-B).LT.0.0) GO TO 3
      IF((C-A).LT.0.0.AND.(D-B).GT.0.0) GO TO 4
      GO TO 9
    3 ANGLE=ANGLE-3.1415926536
      GO TO 9
    4 ANGLE=3.1415926536-ANGLE
    9 CONTINUE
      IF(SIN(ANGLE))5,6,5
    5 CONTINUE
      RLNGTH=(D-B)/SIN(ANGLE)
      GO TO 99
    6 RLNGTH=C-A
   99 CONTINUE
      RETURN
      END
```

```
      SUBROUTINE FBLANG(A,B,C,D,THETA,PHI,PSI)
C  FBLANG MEANS  FOUR BAR LINKAGE ANGLES
C  A,B,C,D ARE THE LENGTHS OF THE LINKAGES
      TOP=A*A+B*B-2*A*B*COS(THETA)
      BOT=SQRT(A*A+B*B-2*A*B*COS(THETA))
      ENDTM=ATAN((B*SIN(THETA))/(A-B*COS(THETA)))
      PHI=ARCOS((TOP+C*C-D*D)/(2*C*BOT))-ENDTM
      PSI=3.14159265-ARCOS((TOP-C*C+D*D)/(2*D*BOT))-ENDTM
      RETURN
      END



      SUBROUTINE FINDXA(A,B,L,D,X,ANGLE)
C  FINDXA CALCS X-COORD AND ANGLE GIVEN A POINT, LENGTH AND Y-COORD
C  NOTE: POINTS FOUND ARE IN QUADRANTS ONE AND FOUR ONLY
      ANGLE=ARSIN((D-B)/L)
      X=A+(L*COS(ANGLE))
      RETURN
      END



      SUBROUTINE FINDXY(A,B,RLNGTH,ANGLE,C,D)
C  FINDXY CALCS COORDS GIVEN A POINT, LENGTH AND ANGLE
C    A,B,C,D ARE THE COORDS OF THE TWO POINTS
      C=A+RLNGTH*COS(ANGLE)
      D=B+RLNGTH*SIN(ANGLE)
      RETURN
      END
```

PRELIMINARY ANALYSIS OF WISHBONE SUSPENSION GEOMETRY

	X-COORD	Y-COORD
1	330.0	419.0
2	262.0	219.0
3	522.0	414.0
4	555.0	207.0
5	0.0	0.0
6	632.0	280.0
7	637.0	0.0

TO CALCULATE FOR BOUNCE 100 AND REBOUND 100

LENGTH OF UPPER ARM IS 192.065

LENGTH OF LOWER ARM IS 293.246

LENGTHS	L12	L13	L24	L34	L35	L36	L37
	211.244	192.065	293.246	209.614	666.243	173.367	429.675

ANGLES	AX12	AX13	AX24	AX34	AX35	AX36	AX37
	-1.899	-0.026	-0.041	-1.413	-2.471	-0.883	-1.300

ANGLES	A435	A436	A437
	-1.058	0.529	0.113

KINGPIN ANGLE = 8.035

SCRUB IS POSITIVE OUTWARDS

K	Y7	RY	RY-Y7	CAMBER	SCRUB	SARM	RATIO
* 0	0.000	35.418	35.418	-1.023	0.000	13887.968	10.901
** 25	25.521	34.152	8.631	-1.280	0.880	3598.710	2.825
** 50	51.705	34.223	-17.483	-1.851	0.691	2067.577	1.623
** 75	78.701	37.287	-41.413	-2.771	-0.571	1413.204	1.109
** 100	106.821	45.553	-61.268	-4.121	-2.866	1031.493	0.810
** 0	0.000	35.418	35.418	-1.023	0.000	13887.968	10.901
** -25	-24.890	36.360	61.250	-1.078	-1.896	6399.862	5.023
** -50	-49.068	34.557	83.625	-1.481	-4.674	2258.269	1.773
** -75	-72.290	25.026	97.315	-2.321	-8.043	1159.415	0.910
*-100	-93.974	-6.404	87.570	-3.786	-11.351	616.642	0.484

APPENDIX 4.1.

CORRESPONDENCE BETWEEN THE UNIVERSITY OF CANTERBURY AND UNIROYAL LIMITED.

P.O. Box 2261
Christchurch
NEW ZEALAND
5 - 7 - 77

Sales Engineering Department
(Powergrip HTD)
Uniroyal Limited
82 Horseferry Road
London SWIP 2AH
ENGLAND

Dear Sir

At the University of Canterbury we are very interested in your company's Powergrip HTD Belts. I am currently carrying out postgraduate work on the design of a prototype electric town car, to replace the interim vehicle currently running at the University. This project has come about from the development of a variable speed AC Motor-drive system, patented by Mr D.J.Byers of our electrical department.

The existing vehicle - using motors rated at 2.2 kW - has two Synchro $\frac{1}{2}$" pitch belts obtained locally. The first, a $1\frac{1}{2}$" belt, ratio 2.4 and the second a 2" belt, ratio 1.85, for each of the two driven wheels.

The prototype vehicle will, with one belt require a ratio of 6.0 or 5.54 (because of the different tyre diameter being used) for which the Synchro $\frac{1}{2}$" pitch belt will be unsuitable, especially with tooth jumping under regenerative braking.

The larger of the two motors that can be used is rated at 4.1 kW at 1500 rpm. However the AC System provides a constant torque of 26 Nm up to 3000 rpm, whereupon the torque drops until the maximum speed of 4500 rpm is reached. The power therefore is 8.2 kW at 3000 rpm. A dropout torque of 2 to 2.5 times the rated torque is available if required. However, the belt will not be subjected to this dropout torque continuously, but only during maximum acceleration, hill climbing etc..

Would an 8 mm pitch 50 mm wide belt with a ratio of 6.0 and pulleys 24/144 at a centre distance of 257.1 mm and belt length of 1280 mm be satisfactory? Is there a pitch size available between 8 mm and 14 mm? Using a borrowed 1974 Powergrip HTD handbook and amendment, and accounting for the additional 0.4 speedup factor (regenerative braking) it appears that an 8 mm pitch 50 mm wide belt may just suffice. If this is the case, could it be run at 4500 rpm on occasions? (1000 rpm corresponds to 19.5 kph; 3000 rpm to 58 kph - speed limit is 50 kph and 4500 rpm to 88 kph - open road speed limit is 80 kph)
Could we expect any trouble with the belt jumping teeth under regenerative braking?

The belt life is not very critical. It would be reasonable, (depending on the belt cost) for a production model, to replace the belts when the tyres are replaced.

The larger pulley of cast iron is unacceptable, as it is unsprung weight on the rear wheel. I understand that filled nylon is the latest development in pulley material, and this would be acceptable. Is your company in a position to make special pulleys ?

We would greatly appreciate any information and aid your company is able to give us on this project (which may lead to production). Looking forward to your reply and to doing business with your company in the future.

Yours faithfully

Edgar Vandendungen

UNIROYAL RTC/SMP

UNIROYAL LIMITED
62 Horseferry Road
London
SW1P 2AH
Telephone: 01-222 5611
Telex: 22814

Registered in Edinburgh
Registration Number 1771
Registered Office: Newbridge, Midlothian, Scotland.

19 July 1977

cc. Mr. A.S. Forgas,
PowerGrip Industries

Edgar Vandendungen, Esq.,
Mechanical Engineering,
P.O. Box 2261,
Christchurch,
New Zealand.

Dear Mr. Vandendungen,

Thank you very much for your letter of July 5 giving details and enclosing photographs of the proto-type electric town car on which you are working.

This is extremely interesting and we will be only too happy to assist you in this project. We have sent a copy of your letter to our Product Engineering Department for study and we hope to have their comments on this in the course of the next few days.

I will therefore, be writing to you again as soon as these comments come to hand.

In the meantime, we would advise that our distributor in South East Asia is:

PowerGrip Industries,
37-39 Lexton Road,
Box Hill,
Victoria 3128,
Australia,

to whom we are sending a copy of your letter as we believe that it will probably be easier and cheaper for you to obtain your pulleys directly from them.

However, we will be able to let you have further information concerning our recommendations for pulleys when we answer your letter in detail.

Assuring you of our best attention at all times.

Yours faithfully,
UNIROYAL LIMITED

R. T. Chalk
European Marketing Manager

UNIROYAL

RTC/SMP

UNIROYAL LIMITED
62 Horseferry Road
London
SW1P 2AH
Telephone: 01-222 5611
Telex: 22814

Registered in Edinburgh
Registration Number 1771
Registered Office: Newbridge, Midlothian, Scotland.

27 July 1977

cc. Mr. A.S. Forgas

Mr. E. Vandendungen,
Mechanical Engineering,
P.O. Box 2261,
Christchurch,
New Zealand.

Dear Mr. Vandendungen,

Further to my letter of July 19 I have now received a reply from our Product Engineering Department concerning the proposed application of our PowerGrip HTD drives to your proto-type electric town car. Details are as follows:

1. Drive specification

Driver pulley	24-8M-50F
Driven pulley	144-8M-50
Max power	4.1 Kw at 1500 rpm
Rated torque	26 nm
Max torque	26 x 2.5 = 65 nm
Max rpm	4,500

2. The 50 mm wide drive which we have recommended will be running well within its limits providing the torque during regenerative braking is not more than approximately 80 nm.

3. We note that the driven pulley is unsprung mass on the rear wheel. The drive will have to be arranged so that the suspension movement does not affect belt tension or alignment of the pulleys, i.e. suspension centre of rotation will have to coincide with driving pulley centre.

4. The drive must be guarded against stones, etc. from the road.

5. It is highly unlikely that the driven pulley could be injection moulded in any sort of plastic to the required tolerances although it could be machined from nylon if a big enough piece could be obtained or a fabric filled phenolic, particularly if this were graphite loaded could also be used. Alternatively, an aluminium alloy pulley hard anodised would probably be the easiest and most suitable material for the driven pulley with mild steel being used for the drive pulley.

Cont. /...

6. If it is possible to consider a two stage drive reduction, this would give a more compact drive using 26 and 64 groove pulleys on both primary and secondary drives. A 50 mm wide belt would be used on the primary drive and an 85 mm wide belt would be needed for the secondary drive. The maximum pulley OD of 161.6 mm would then be a more reasonable size for moulding.

We hope that these comments will be of assistance to you and we will be most interested to learn of future developments with this electric vehicle. However, if there is any further information we can give you on this matter, please do not hesitate to contact us.

Yours sincerely,

R. T. Chalk

Edgar Vandendungen
P.O. Box 2261
Christchurch
NEW ZEALAND
21 Sept. 77

Mr R.T. Chalk
Uniroyal Limited
62 Horseferry Road
London SW1P 2AH
ENGLAND

Dear Mr Chalk

Thank you very much for your letters dated 19 and 27 July. We found the information very helpful and the offer of assistance most gratifying. We intend to use your companies belts for the electric town car and have altered the design width of the chassis and bodywork to suit the narrower 50 mm belts.

The driven pulley, as you point out, is in fact unsprung mass on the rear wheels, as is the motor and driver pulley, thus ensuring a fixed pulley-centre distance. However the effective unsprung mass of the motor is approxiamately 40% of the actual mass, and when calculated, the unsprung mass is approxiamately 70% that of a Ford Escort live axle, for instance.

Of some concern is the amount of flex of the rear suspension arm that can be tolerated, ie: how much angular misallignment between one pulley and the other can be tolerated before the belt will start running off. Do you have any figures on this ?

Taking your advice, we now have a local company that is prepared to hard anodize our aluminium driven pulleys. Would it be possible, when the time comes, for us to machine the driven pulleys in Christchurch to your specifications, or would it be advisable for your distributor in Australia to do the required machining.

A two stage drive reduction was considered at one time, but was rejected on the grounds that it required an extra belt and two pulleys, plus the extra layshaft and bearing on each end, as well

as taking up extra space. The motor located near the pivot on the swingarm, appears the best compromise.

Thank you once again for your assistance. We look forward to hearing from you, and working with your company in the future.

Yours faithfully

Edgar Vandendungen.

UNIROYAL LIMITED
62 Horseferry Road
London
SW1P 2AH
Telephone: 01-222 5611
Telex: 22814

Registered in Edinburgh
Registration Number 1771
Registered Office: Newbridge, Midlothian, Scotland.

c.c. Mr. A.S. Forgas,
PowerGrip Industries

Mr. Edgar Vandendungen,
Mechanical Engineering,
P.O. Box 2261,
Christchurch,
NEW ZEALAND.

Our Ref: RTC/MYB

20 October 1977

Dear Mr. Vandendungen,

Thank you for your letter of 21 September 1977, raising points concerning potential angular misalignment on the proposed PowerGrip HTD Drive, which you are considering for your electric car.

I have raised this matter with our Product Engineering Department, and their comments are as follows:-

"My misgivings about mounting the driven pulley as unsprung weight on the back axle were due mainly to the possibility of drastic variations in belt tension. This could be due either to the centres of rotation of the axle and driving pulley not coinciding, or to compliance in the suspension bushes permitting longitudinal movement of the axle. The compliance of the belt will almost certainly be lower than that of rubber bushes, which would result in the belt acting as a sort of radius arm.

The effect of angular misalignment is to elongate the outside edges of the belt with respect to the centre by an amount equal to:

$$\sqrt{\left(W \sin \frac{\theta}{2}\right)^2 + C^2} - C$$

where W = Belt Width
θ = Misalignment
C = Belt Span between Pulleys

This is not too serious if C is reasonably large and W reasonably small.

These comments notwithstanding, unorthodox things often work, but it is better to be aware of possible difficulties".

cont .....

I hope that these comments will be of help to you. However, I should be most interested to learn of your developments, and if there is any further information you may require at any time, please do not hesitate to let me know.

Yours sincerely,

R. T. Chalk,
European Marketing Manager,
Power Transmission Division.

May 10, 1978

Power Grip Industries,
37-39 Lexton Rd.,
Box Hill,
Victoria 3128,
AUSTRALIA.

Dear Sirs,

University of Canterbury, Electric Vehicle Project

I am writing to follow up some correspondence between Mr. R.T. Chalk of Uniroyal, Ltd., London and a student whom I supervise, Mr. E. Vandendungen. Mr. Chalk gave me your address and said he would send you a copy of the correspondence. I confirmed our continuing interest to Paykel Bros.Ltd., Christchurch on October 28, 1977, in response to a query from you as I understand it.

The design of our Mk.II vehicle is well under way, and we are now in a position to order belts and pulleys for it. There will be only one car at first but there is the possibility of a very small production run of evaluation vehicles to the same design. Each car will have two belt drives as follows:

Belts - Uniroyal Powergrip HTD, pitch 8 mm
width 50 mm
length 1280 mm

Drive N pulleys - 144-8M-50, hard anodized aluminium
Drive R pulleys - 26-8M-50F or 28-8M-50F, iron or steel

While the belts and driver pulleys are standard, we are interested in means of obtaining suitable lightweight driven pulleys to mount beside the rear wheels. The form required will be a simple toothed ring with a central internal attachment flange, as sketched below. Three are required, to include a spare in case of trouble.

Would you please advise on the following:

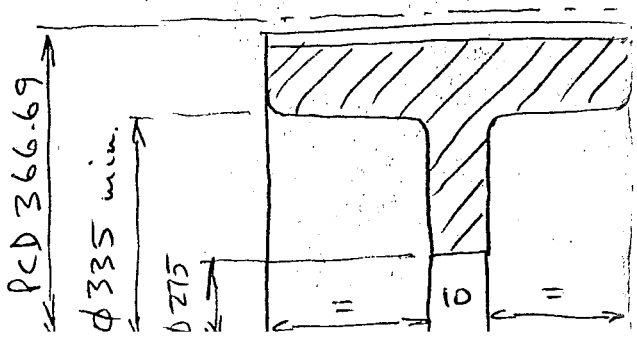
1. Price and availability for the belts and the optional driver pulleys as listed above.
2. Price and delivery date for three rings as sketched for the driven pulleys, fully machined and anodised.
3. The possibility of your supplying a tooth profile cutter and machining instructions so we may arrange our own driven pulleys.

Looking forward to your reply.

Yours sincerely,

R.T.C. Harman
Senior Lecturer in Mechanical Engineering

c.c. Mr. T.J. Whall, Paykel Bros. Ltd., P.O. Box 382, Christchurch



POWERGRIP INDUSTRIES (AUSTRALIA) PTY. LTD.

POWER TRANSMISSION & CONVEYOR BELT MANUFACTURERS
IMPORTERS & EXPORTERS

37-39 LEXTON ROAD, BOX HILL, VIC., 3128, AUSTRALIA.

P.O. BOX 223, BOX HILL, 3128. Phone: 89 0344 (7 lines). Telex: AA33610

Distributors of
Uniroyal, Powergrip Timing Belt, U.S.A.

Telegrams & Cables: "POWERBELT," MELBOURNE.
IN N.S.W.: Powergrip (N.S.W.) Pty. Ltd., 51 Crystal Street, Petersham, 2049.
IN QLD.: Powergrip Industries (Qld.) Pty. Ltd., 237 Montague Road, West End, Queensland, 4101.
Distributors: South Australia, Western Australia, Tasmania.

RS/BF

8th June, 1978.

University of Canterbury,
Department of Mechanical Engineering,
CHRISTCHURCH. 1.
NEW ZEALAND.

For Attention: Mr. R.T.C. Harman,
Senior Lecturer in Mechanical Engineering.

Dear Sir,

We are writing to apologize for delays in our quotation for the supply of Uniroyal Powergrip HTD Drives. The problem at this end has been finding a suitable supplier for the hard anodized aluminium from which the driven pulleys are to be machined. We assure you, however, as soon as we are able to obtain a price for this material our quotation for the drives will be forwarded to our New Zealand Agents, Paykel Bros. Ltd. who in turn will be in touch with you.

Yours faithfully,
POWERGRIP INDUSTRIES (AUST.) PTY.LTD.

R. SMITH,
PURCHASING OFFICER.

c.c. Mr. T.J. Whall,
Paykel Bros. Ltd.,
P.O. Box 382,
CHRISTCHURCH. N.Z.

June 14, 1978

Powergrip Industries (Australia) Pty.Ltd.,
P.O. Box 223,
Box Hill,
Victoria 3128,
AUSTRALIA.

Attention: Mr. R. Smith, Purchasing Officer

Dear Mr. Smith,

Many thanks for your letter of 8th. June. I am replying quickly because your letter implies that the material of the pulleys would be hard anodized before machining. In fact, they need to be machined in an as cast state, and then anodized to harden the wearing surfaces of the teeth. My apologies if I have misunderstood you on this point.

I am not sure which particular aluminium alloy is required for this process, but it is used and hard anodized here for the manufacture of tyre moulds. If necessary, we could have much of the manufacture completed locally.

We are hoping to have this material to hand in early August. The pulley sizes required are still as requested but the belts will change to the next longer size, becoming 1360 mm long.

Hoping that this gives you adequate time to obtain and make the required parts.

Yours faithfully,

R.T.C. Harman
Senior Lecturer in Mechanical Engineering

c.c. Mr. T.J. Whall,
Paykel Bros. Ltd.,
P.O. Box 382,
CHRISTCHURCH

25% duty.

171

POWERGRIP INDUSTRIES

POWER TRANSMISSION & CONVEYOR BELT MANUFACTURERS
IMPORTERS & EXPORTERS

37-39 LEXTON ROAD, BOX HILL, VIC., 3128, AUSTRALIA
P.O. BOX 223, BOX HILL, 3128, AUSTRALIA. Phone: 89 0344 (7 lines)
Telegrams & Cables: "POWERBELT", MELBOURNE, AUSTRALIA.

RD/BF

LCF 1.3.2

Paykel Bros. Ltd.,
P.O. Box 5046,
AUCKLAND. NEW ZEALAND.

17th July, 1978.

Attention: Mr. B. Mitchell, Manager, Transmission Division.

Quotation No. 1993/Mr.R.T.C.Harman, Dept.Mech.Eng.
University of Canterbury, Christchurch.N.Z.

Dear Barry,

Enclosed is a letter received from Mr. R.T.C.Harman of the Department of Mechanical Engineering, University of Canterbury, Christchurch. Further to this letter we have much pleasure in quoting the following.

SACC
84420
84.63029

Item 1. 3 - 144-8M-50 Aluminium Hard Anodized HTD Pulleys, NZ @ $645.97 NETT Each.
* $484.48

Item 2. 3 - 28-8M-50F Steel HTD Pulleys, @ $63.48 NETT Each.
* $47.61

Item 3. 1280-8M-50 ------ HTD Timing Belts, @ $142.42 NETT Each. NIL
40 14 001

Item 4. 3 - 1440-8M-50 HTD Timing Belts, @ $147.02 NETT Each. NIL
40.14.001

IE.

ALTERNATIVE TO ITEM 1.

3 - 144-8M-50 HTD Pulley, Steel Fabrication Lightweight, @ $527.18 NETT Each.
* $ 395.39

1. Delivery of HTD Pulleys - 8 weeks.
2. HTD Timing Belts - 20 weeks Sea Freight.
3. " 10 " Air Freight.

Letter dated June 14th refers to a belt size of 1360mm long, but our next standard size is 1440mm.

As HTD is a patented product we cannot supply HTD Cutters - Uniroyal will prosecute anyone trying to duplicate the cutting of HTD Timing Pulleys.

Looking forward to an early reply,

Yours sincerely,
POWERGRIP INDUSTRIES (AUST.)PTY.LTD.
R.DALE, SALES MANAGER.

* LESS 25% DUTY.

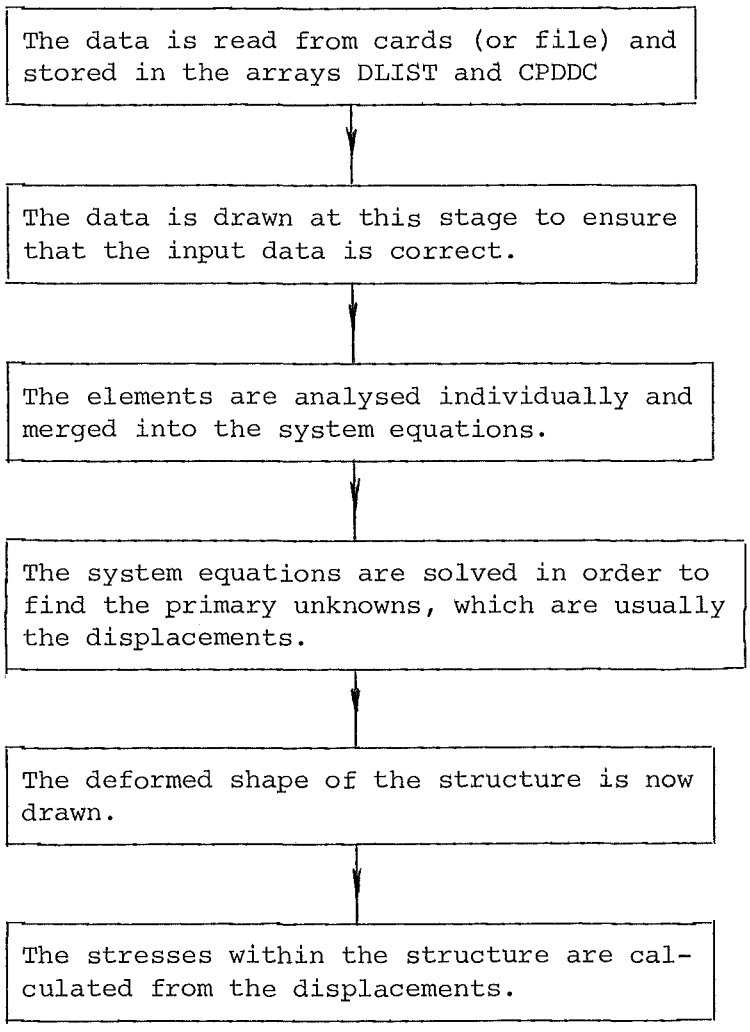
BELTS: DUTY NOT APPLICABLE.

less machining time?

APPENDIX 4.2

FINITE ELEMENT ANALYSIS USING PAFEC 75 ON A B6700 COMPUTER. APPLICABLE TO THE REAR SUSPENSION ARM OF AN ELECTRIC TOWN CAR

FLOWCHART



Discussion. The finite element program PAFEC, which stands for Program for Automatic Finite Element Calculations, is run in 10 separate steps called phases. Each step has a different function to perform which is summarised below.

Phase 1	Data input and expansion.
Phase 2	Mesh generation using Pafblocks.
Phase 3	Draw finite element mesh.
Phase 4	Allocates degrees of freedom to nodes.
Phase 5	Draw degrees of freedom.
Phase 6	Generates element stiffnesses.
Phase 7	Merging and solution routines.
Phase 8	Displaced shape or temperature contour plots.
Phase 9	Calculates element stresses.
Phase 10	Stress vector or contour plots.

Data for PAFEC is divided into separate blocks called Modules (ref. PAFEC 75 EASIDATA MANUAL DEC. 1977). The type of calculation to be performed (e.g. static or dynamic analysis) and the number and type of Phases to be run may be specified in the CONTROL MODULE which is one module of the data file.

The program as set up uses the control module and phases one, three, four, six, seven, eight and nine. This is reflected in the flow chart above. The drawings from phases three and eight are included in Chapter 4. Phase one interprets and lists the input data. The output from phase four is a list of all 408 degrees of freedom for the 68 nodes, plus the addresses and tags for each degree of freedom. The output from phase six is a list of the group number, element type, property number and material number for the 83 elements.

The output from phase seven tabulates the reactions at the constrained nodes, the trailing arm pivots, for each of the three load cases analysed. It also tabulates, separately for each load case, the coordinates for each node and the displacement of each of the degrees-of-freedom, that is, the x, y and z displacements and the rotations about the x, y and z local axes. The output from phase nine lists all the stresses for each element and/or each node of each element for each load case. For the 3-node triangular elements, the maximum stress, the minimum stress and the angle between them for each of the positive surface, the neutral surface and the negative surface for each of the load cases is listed, while for the four node elements this is done at each node of each element for each load case. For beam elements, the axial force and shear force in both directions and the torsional moment and bending moment in both directions is listed for all load cases.

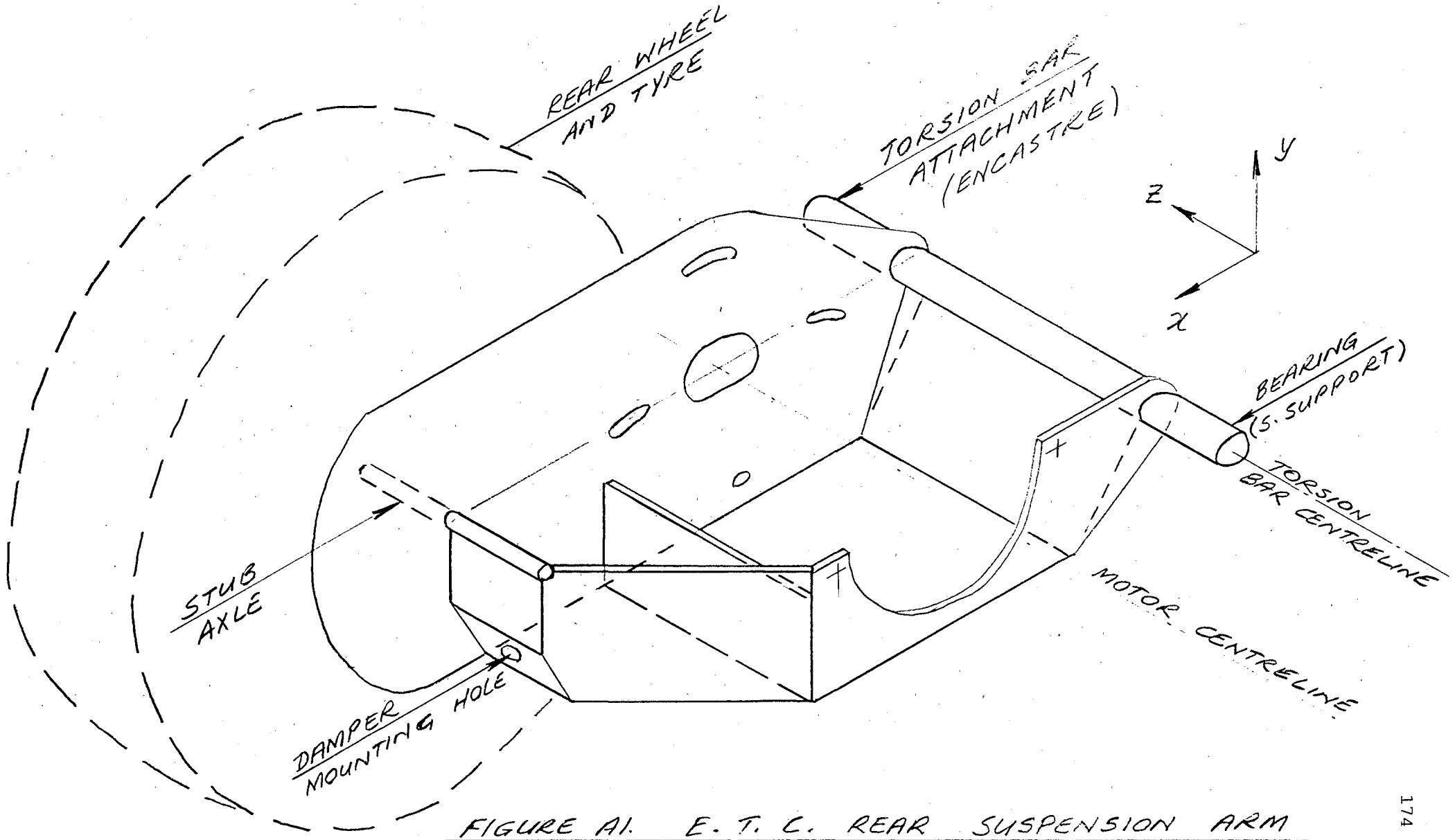


FIGURE A1. E. T. C. REAR SUSPENSION ARM

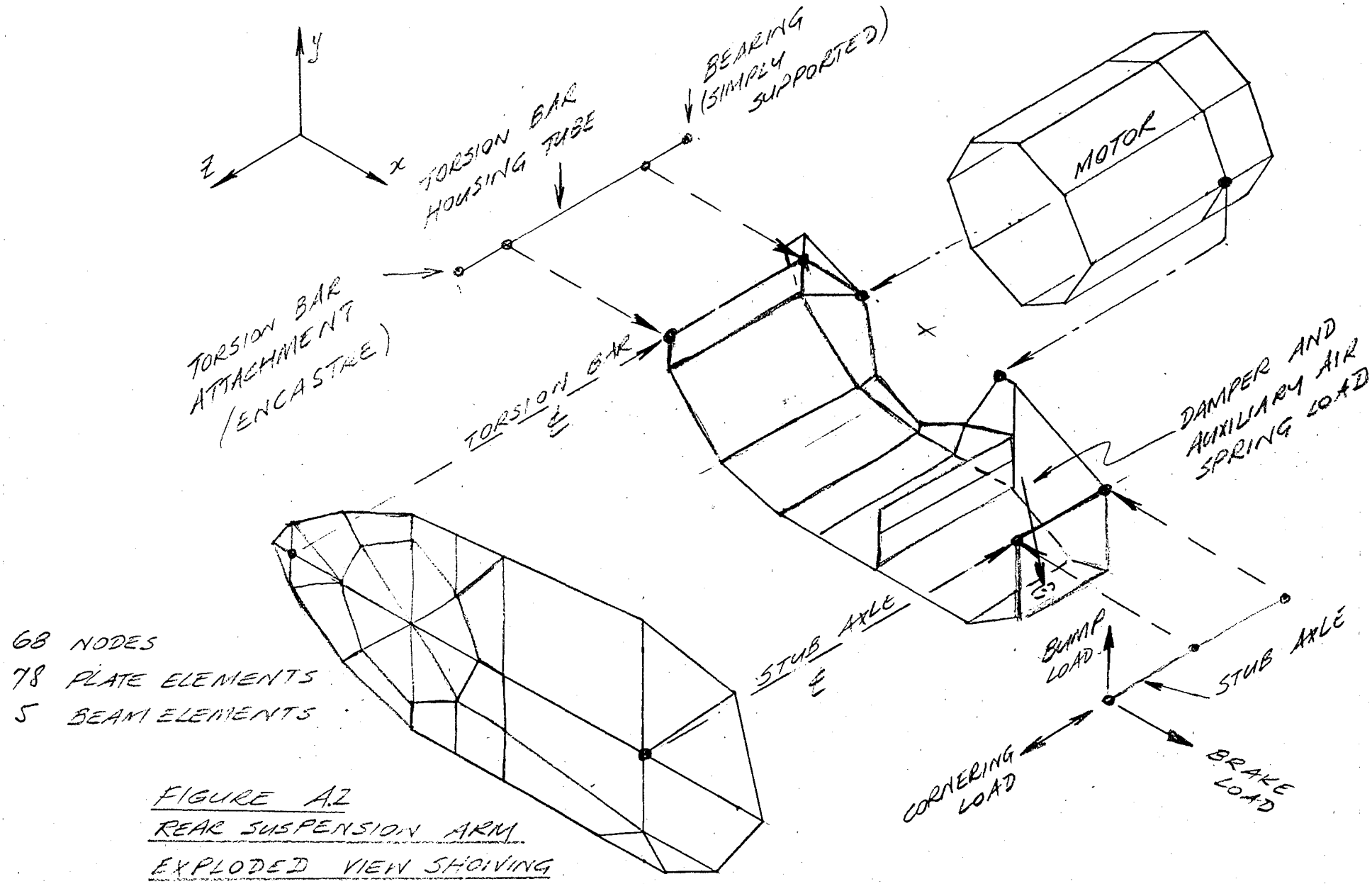


FIGURE A.2
REAR SUSPENSION ARM
EXPLODED VIEW SHOWING
FINITE ELEMENTS.

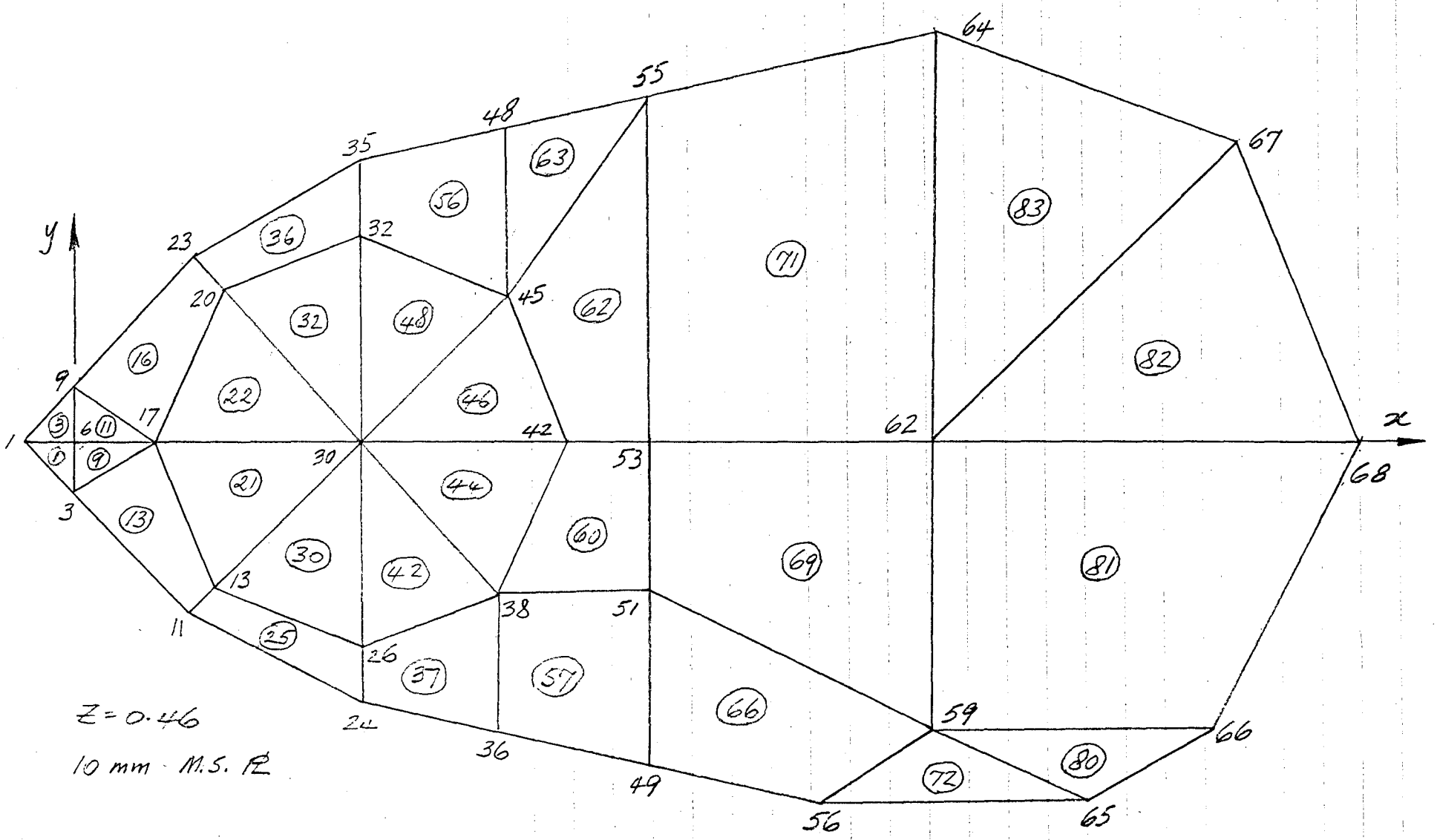


FIGURE A.3. FINITE ELEMENT NODES AND ELEMENTS FOR OUTER SIDE PLATE.

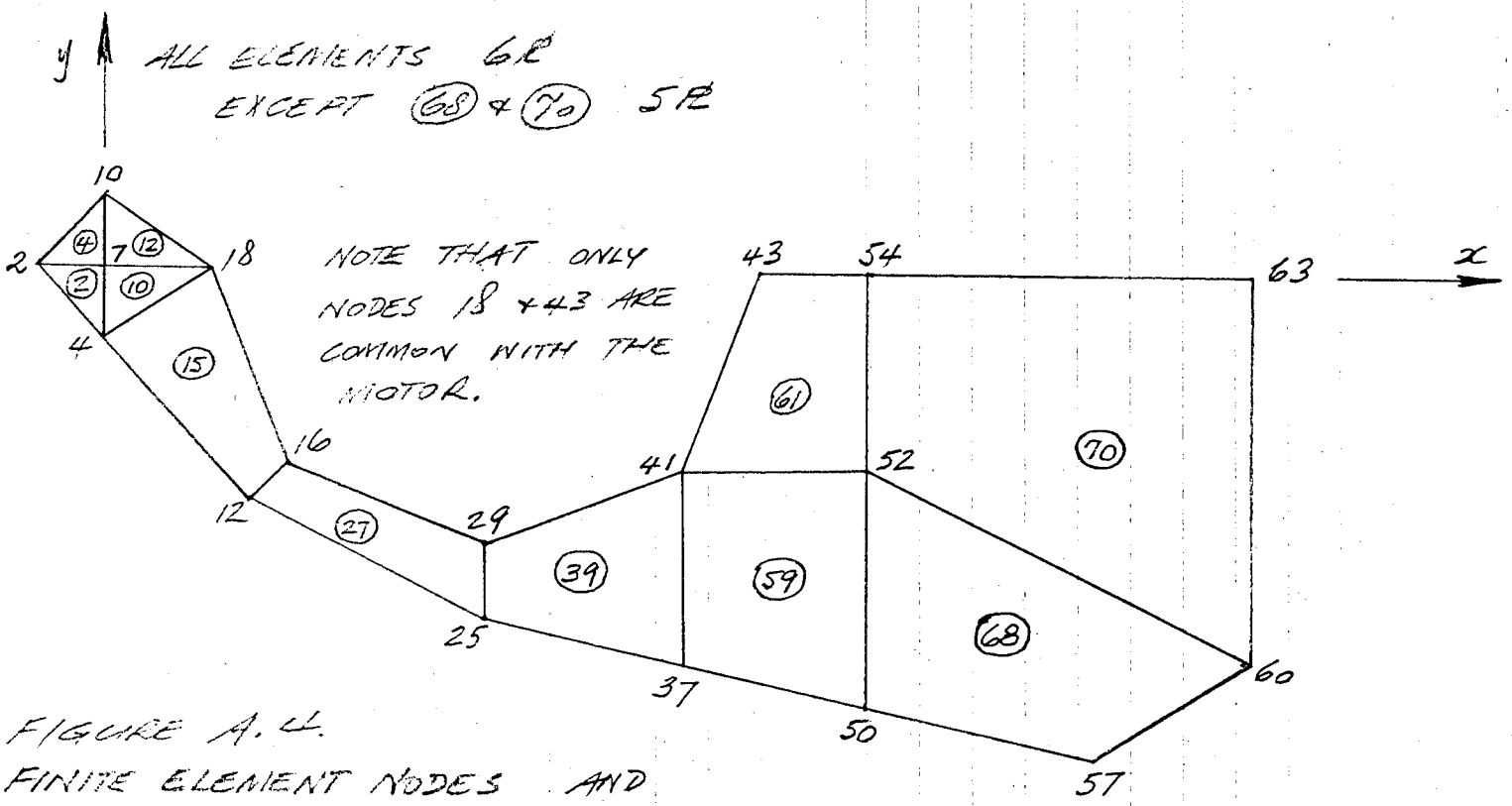


FIGURE A.4.
FINITE ELEMENT NODES AND ELEMENTS FOR INNER SIDE PLATE

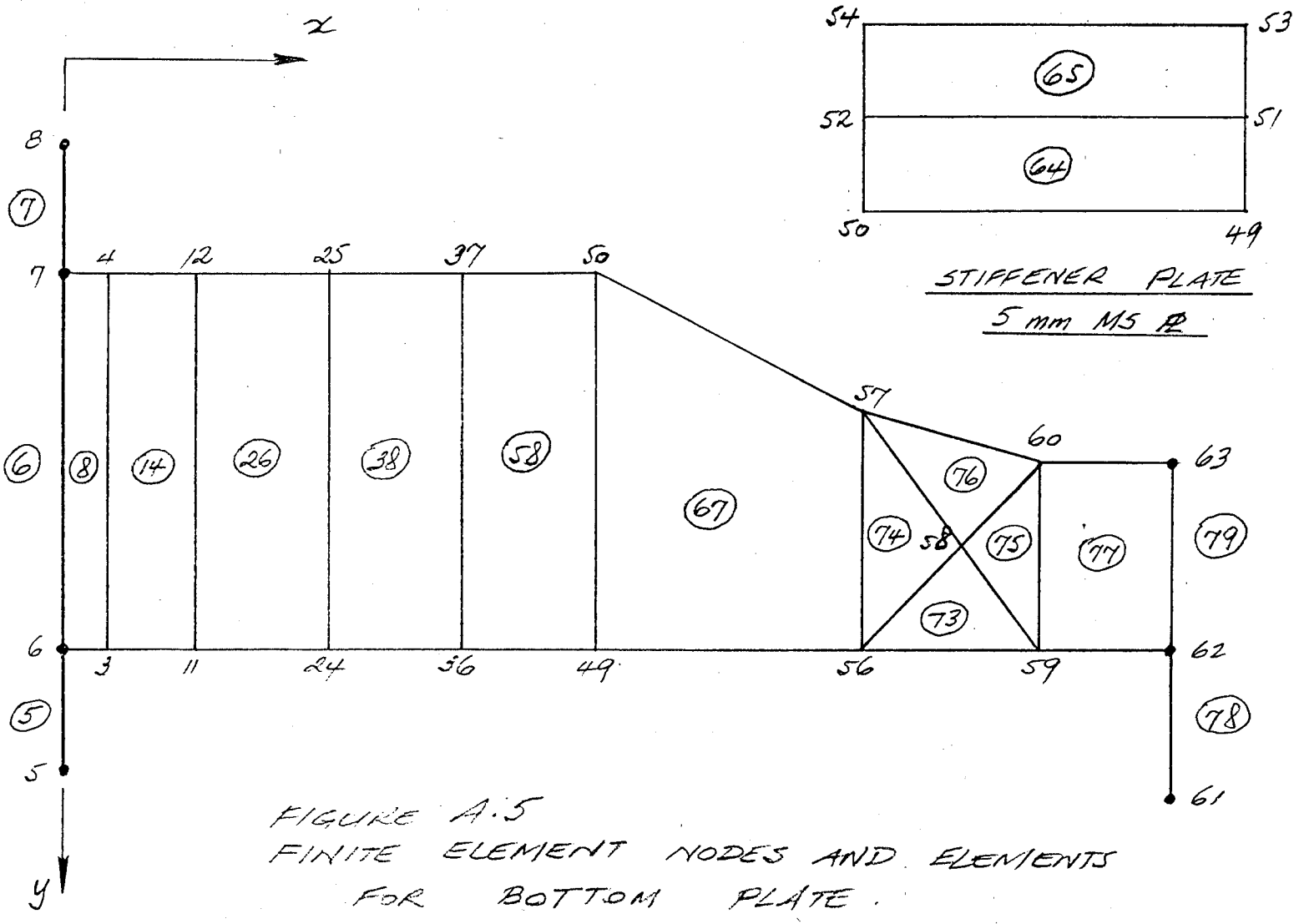


FIGURE A.5
FINITE ELEMENT NODES AND ELEMENTS
FOR BOTTOM PLATE.
6 mm MS

NOTE THAT ONLY NODES 18 AND #3 ARE COMMON WITH INNER SIDE PLATE.

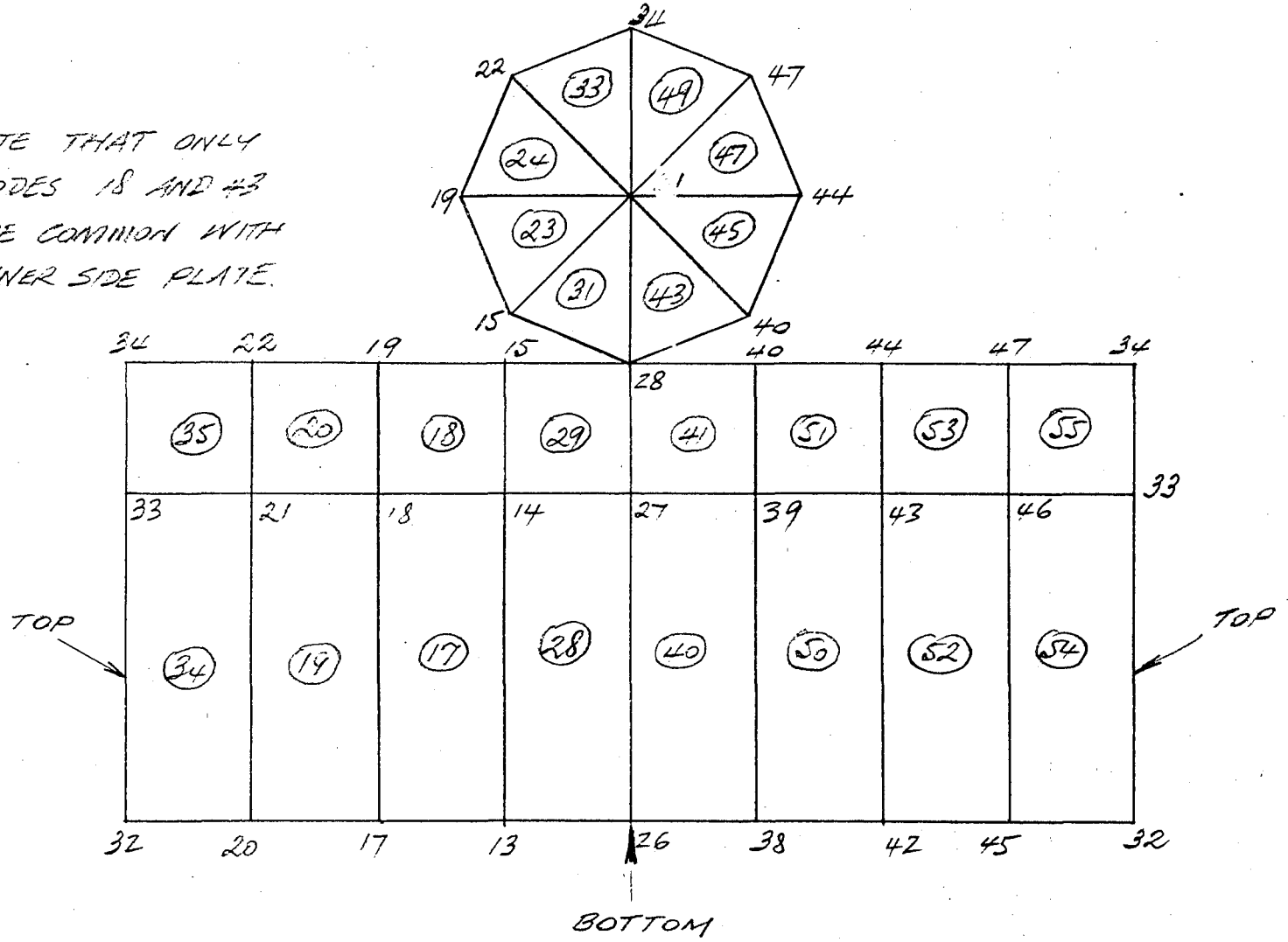


FIGURE A.6
FINITE ELEMENT NODE AND ELEMENTS FOR MOTOR (SHOWN DEVELOPED)

Data Preparation

Figure A.1 illustrates the rear suspension arm, with the motor omitted for clarity. Drawings 7937/1 and 7956 in Appendix 6.1 provide the detail dimensions. Figure A.2 illustrates the rear suspension arm in exploded view, showing the finite elements, while Figures A.3 to A.6 inclusive illustrate the mode numbers and element numbers, prepared in accordance with the appropriate PAFEC 75 Manuals. The list of FINITE ELEMENT DATA is based on Figures A.3 to A.6 with element types, properties and topologies and other data in accordance with the appropriate PAFEC 75 Manuals.

FINITE ELEMENT DATA

Node numbers not punched, assumed consecutive.

TITLE*ETC*REAR*SUSPENSION*STRESS*ANALYSIS NODES

	x	y	z
1	-.02	0	.46
2	-.02	0	.22
3	0	-.02	.46
4	0	-.02	.22
5	0	0	.54
6	0	0	.46
7	0	0	.22
8	0	0	.075
9	0	.02	.46
10	0	.02	.22
11	.067	-.092	.46
12	.067	-.092	.22
13	.082	-.078	.46
14	.082	-.078	.22
15	.082	-.078	.145
16	.078	-.082	.22
17	.05	0	.46
18	.05	0	.22
19	.05	0	.145
20	.082	.078	.46
21	.082	.078	.22
22	.082	.078	.145
23	.067	.092	.46

TITLE*ETC*REAR*SUSPENSION*STRESS*ANALYSIS NODES

24	.16	-.13	.46
25	.16	-.13	.22
26	.16	-.11	.46
27	.16	-.11	.22
28	.16	-.11	.145
29	.16	-.115	.22
30	.16	0	.46
31	.16	0	.145
32	.16	.11	.46
33	.16	.11	.22
34	.16	.11	.145
35	.16	.13	.46
36	.238	-.145	.46
37	.238	-.145	.22
38	.238	-.078	.46
39	.238	-.078	.22
40	.238	-.078	.145
41	.242	-.082	.22
42	.270	0	.46
43	.270	0	.22
44	.270	0	.145
45	.238	.078	.46
46	.238	.078	.22
47	.238	.078	.145
48	.238	.145	.46
49	.33	-.168	.46
50	.33	-.168	.22
51	.33	-.08	.46
52	.33	-.08	.22
53	.33	0	.46
54	.33	0	.22
55	.33	.160	.46
56	.45	-.195	.46
57	.45	-.195	.315
58	.48	-.175	.4
59	.51	-.155	.46
60	.51	-.155	.36
61	.51	0	.62

TITLE*ETC*REAR*SUSPENSION*STRESS*ANALYSIS NODES

62	.51	0	.46
63	.51	0	.36
64	.51	.195	.46
65	.57	-.195	.46
66	.628	-.138	.46
67	.648	.138	.46
68	.705	0	.46

ELEMENTS

PROP = 1

ELEMENT. TYPE = 41320

NUMBER	TOPOLOGY
1	1, 3, 6
3	1, 6, 9
9	3, 17, 6
11	6, 17, 9
21	13, 30, 17
22	17, 30, 20
30	13, 26, 30
32	20, 30, 32
42	26, 38, 30
44	38, 42, 30
46	30, 42, 45
48	30, 45, 48
63	45, 55, 48
72	56, 65, 59
80	65, 66, 59
82	62, 68, 67
83	62, 67, 64

ELEMENT. TYPE = 44200

NUMBER.	TOPOLOGY
13	3, 11, 17, 13
16	17, 20, 9, 23
25	11, 24, 13, 26
36	20, 32, 23, 35
37	24, 36, 26, 38
56	32, 45, 35, 48
57	36, 49, 38, 51
60	38, 51, 42, 53
62	42, 53, 45, 55
66	49, 56, 51, 59
69	51, 59, 53, 62
71	53, 62, 55, 64
81	59, 66, 62, 68

ELEMENT. TYPE = 41320

PROP = 2

NUMBER	TOPOLOGY
73	56, 59, 58
74	56, 58, 57
75	58, 59, 60
76	58, 60, 57

ELEMENT. TYPE = 41320

PROP = 3

NUMBER	TOPOLOGY
2	2, 4, 7
4	2, 7, 10
10	4, 18, 7
12	7, 18, 10

ELEMENT. TYPE = 41320

PROP = 4

NUMBER	TOPOLOGY
23	15, 31, 19
24	19, 31, 22
31	15, 28, 31
33	22, 31, 34
43	28, 40, 31
45	31, 40, 44
47	31, 44, 47
49	31, 47, 34

ELEMENT. TYPE = 44200

PROP = 2

NUMBER	TOPOLOGY
25	24, 25, 11, 12
38	36, 37, 24, 25
58	49, 50, 36, 37
64	51, 52, 49, 50
65	53, 54, 51, 52
67	56, 57, 49, 50
68	50, 57, 52, 60
70	52, 60, 54, 63
77	62, 63, 59, 60

ELEMENT. TYPE = 44200

PROP = 4

NUMBER	TOPOLOGY
17	13, 14, 17, 18
18	14, 15, 18, 19
19	17, 18, 20, 21
20	18, 19, 21, 22
28	26, 27, 13, 14
29	27, 28, 14, 15
34	20, 21, 32, 33
35	21, 22, 33, 34

NUMBER	TOPOLOGY
40	38, 39, 26, 27
41	39, 40, 27, 28
50	42, 43, 38, 39
51	43, 44, 39, 40
52	45, 46, 42, 43
53	46, 47, 43, 44
54	32, 33, 45, 46
55	33, 34, 46, 47

ELEMENT. TYPE = 44200

PROP = 3

NUMBER	TOPOLOGY
8	3, 4, 6, 7
14	11, 12, 3, 4
15	4, 12, 18, 16
27	12, 25, 16, 29
39	25, 37, 29, 41
59	37, 50, 41, 52
61	41, 52, 43, 54

ELEMENT. TYPE = 34000

NUMBER,	PROP,	TOPOLOGY
5	3	5, 6
6	3	6, 7
7	3	6, 8
78	4	61, 62
79	3	62, 63

PLATES.AND.SHELLS

PLATE.NUMBER,	MATE.NUMBER,	THICK
1	1	.01
2	1	.005
3	1	.006
4	1	.002

BEAMS

SECT.NUMB,MATE.NUMB,IYY,IZZ,AXIS.NUMB,TORS.CONST.AREA

3, 1, 1.701E-7, 1.701E-7, 3, 3.402E-7, 7.278E-4

4, 1, 3.976E-8, 3.976E-8, 3, 7.958E-8, 7.069 E-4.

LOADS

CASE.OF.LOAD, NODE. NUMB, DIRECTN. OF LOAD, VALUE.OF.LOAD.

1	61	2	2658
2	61	2	14752
2	58	1	10385
2	58	2	-10385
3	61	1	6557
3	61	2	9835
3	61	3	-6557
3	58	1	5468
3	58	2	-5468

ENCASTRE

LIST.OF.NODES

5

SIMPLE.SUPPORTS

LIST.OF.NODES

8

STRESS.ELEMENT

START, FINISH

1	16
21,	22
25,	27
30,	30
32,	32
36,	39
42,	42
44,	44

```
46,     46
58,     48
56,     83

END.OF.DATA
```

Load Cases

Load case 1 is the static 1 g loading configuration, assuming a vehicle mass of 1151 kg (2 persons) with a/L = 0.471, so that individual rear wheel loading is 2658 N acting at node 61 vertically upward, that is, in the y-direction (direction 2).

Load case 2 is a 4.5 g bump with 4 persons assuming 1281 kg and a/L = 0.522. The load taken by the torsion bar is nominally P/G_2 = 1.643 from the interconnected suspension analysis.

Rear torsion bar load = 1.643 x static	=	4367
Bump stop/damper/auxiliary spring	=	10385
Total 4.5 g wheel load	=	14752 N

so that;

	N	Direction
Node 61	14752	2
Node 58	-10385	2
	10385	1

Load case 3 is a 3 g bump, a 2 g corner and 2 g braking for 4 persons, all combined.

Bump 3 g.

Rear torsion bar load	4367
Bump stop/damper/auxiliary spring	5468
Total 3 g wheel load	9835 N

Corner/Braking 2 g = 6557 N, so that

	N	Direction	
Node 61	6557	1	brake
	9835	2	bump
	-6557	3	corner
Node 58	5468	1	
	-5468	2	

Node 61 is the end of the stub axle, and node 58 is the damper/ auxiliary spring mounting hole, see Figures A1 and A5.

Suspense Arm Twist

With reference to the displacements/rotations printout from phase seven, the angular misalignment between the motor shaft and the end of the stub axle in rear elevation, is the difference in angle PHIX, for nodes 61 (Figure A.5) and 30 (Figure A.3).

Consider load case 2, the 4.5 g bump condition with 4 persons. The angles, dividing by 100 as per printout, are

Node 61, PHIX = -0.032593
Node 30, PHIX =-(-0.002825)
-0.029768 radians

The angular misalignment or twist is 1.7 degrees.

Remove Metal to Save Unsprung Weight

That section of the suspension arm comprising nodes 48, 55, 64, 67, 68, 66 and 65 Figure A.3 was originally provided to offer protection for aluminium belt pulley and provide a basis for a belt guard. Chain is now being used and the same level of protection is not necessary, it is therefore proposed to cut off part of the arm.

The suspected understressed elements, Figure A.3. are 63, 71, 83, 82, 81 and 80. The triangular elements 63, 83, 82 and 80 are stressed per element whereas the 4-noded elements are stressed per node, the relevant nodes are 55, 64, 68 and 66. Two printout pages from phase nine are included, from these and other relevant pages the following stresses MN/m^2 are read.

Triangular Element	Load Case 1	2	3
63	+11, -15	+54, -74	+40, -55
83	+ 9, -10	+53, -55	+30, -30
82	+ 2, - 3	+13, -17	+ 5, - 9
80	+ 2, - 6	+41, -33	+16, - 8

4-Node Element	Node	Load Case 1	2	3
71	55	+ 8, -12	+42, -61	+19, -26
71	64	+ 7, -13	+45, -67	+17, -40
81	68	+ 5, - 5	+25, -27	+10, -14
81	66	+10, -13	+64, -83	+41, -43

In the static condition, load case 1, the stresses are insignificant, and even at 4.5 g bump, load case 2, which is an extreme condition, the maximum stress is only 83 MPa compression. Material can be removed without any detrimental effect.

SCALED COORDINATES — DIVIDE BY 10.000

SCALED DISPLACEMENTS/ROTATIONS FOR LOAD CASE 1 — DIVIDE BY 100.00
* INDICATES A CONSTRAINT HAS BEEN APPLIED — NODES 1 THRU 68

COORDINATES X	Y	Z	NODE NO.	DISPLACEMENTS UX	UY	UZ	ROTATIONS PHIX	PHIY	PHIZ
-0.2000	0.0000	4.6000	1	-0.0003	-0.0297	0.0004	0.0033	0.0421	0.0000
-0.2000	0.0000	2.2000	2	0.0001	-0.0276	0.0002	-0.0219	0.0125	0.0000
0.0000	-0.2000	4.6000	3	0.0288	-0.0005	0.0002	-0.0014	0.0413	1.9391
0.0000	-0.2000	2.2000	4	0.0243	-0.0025	0.0004	-0.0069	0.0430	1.4184
0.0000	0.0000	5.4000	5	*	*	*	*	*	*
0.0000	0.0000	4.6000	6	-0.0003	0.0005	-0.0001	0.0020	0.0079	0.3458
0.0000	0.0000	2.2000	7	0.0001	-0.0023	0.0001	0.0005	-0.0011	0.7850
0.0000	0.0000	0.7500	8	*	*	*	0.0230	0.0011	0.7850
0.0000	0.2000	4.6000	9	-0.0315	0.0006	-0.0000	-0.0279	0.0719	0.0000
0.0000	0.2000	2.2000	10	-0.0262	-0.0022	0.0002	0.0001	0.0005	0.0000
0.6700	-0.9200	4.6000	11	0.1406	0.1043	0.0003	-0.0647	0.0431	1.4285
0.6700	-0.9200	2.2000	12	0.1236	0.0890	0.0005	-0.1100	0.0695	1.3956
0.8200	-0.7800	4.6000	13	0.1189	0.1274	-0.0011	-0.0678	0.0270	1.5422
0.8200	-0.7800	2.2000	14	0.1067	0.1092	-0.0016	-0.0265	0.0491	1.4738
0.8200	-0.7800	1.4500	15	0.1058	0.1087	-0.0023	-0.0134	0.0219	1.5265
0.7800	-0.8200	2.2000	16	0.1091	0.1053	-0.0013	-0.0990	0.0632	0.0000
0.5000	0.0000	4.6000	17	-0.0012	0.0777	-0.0045	-0.0444	0.0903	1.5416
0.5000	0.0000	2.2000	18	-0.0016	0.0644	-0.0043	-0.0036	0.1354	1.5396
0.5000	0.0000	1.4500	19	-0.0140	0.0598	-0.0032	-0.0621	0.1115	1.5386
0.8200	0.7800	4.6000	20	-0.1216	0.1272	-0.0096	-0.0359	0.0565	1.5504
0.8200	0.7800	2.2000	21	-0.1338	0.1180	-0.0107	-0.0959	0.0403	1.6147
0.8200	0.7800	1.4500	22	-0.1344	0.1095	-0.0115	-0.0844	0.0106	1.5547
0.6700	0.9200	4.6000	23	-0.1434	0.1041	-0.0092	-0.0410	0.0780	0.0000
1.6000	-1.3000	4.6000	24	0.1993	0.2481	-0.0033	0.0129	0.0584	1.5727
1.6000	-1.3000	2.2000	25	0.1817	0.2300	-0.0034	-0.0752	0.0709	1.5640
1.6000	-1.1000	4.6000	26	0.1685	0.2481	-0.0032	-0.0007	0.0490	1.5500
1.6000	-1.1000	2.2000	27	0.1581	0.2345	-0.0030	-0.0002	0.0302	1.5689
1.6000	-1.1000	1.4500	28	0.1549	0.2289	-0.0028	-0.0090	0.0436	1.5481
1.6000	-1.1500	2.2000	29	0.1587	0.2301	-0.0046	-0.0797	0.0655	0.0000
1.6000	0.0000	4.6000	30	-0.0013	0.2477	-0.0077	-0.0032	0.0115	0.0000
1.6000	0.0000	1.4500	31	-0.0145	0.2291	-0.0099	-0.0577	0.0503	0.0000
1.6000	1.1000	4.6000	32	-0.1712	0.2477	-0.0156	-0.0826	0.0426	1.5375
1.6000	1.1000	2.2000	33	-0.1911	0.2333	-0.0158	-0.0635	0.0122	1.4930
1.6000	1.1000	1.4500	34	-0.1839	0.2293	-0.0157	-0.0539	0.0415	1.5274
1.6000	1.3000	4.6000	35	-0.2020	0.2477	-0.0172	-0.0883	0.0527	0.0000
2.3800	-1.4500	4.6000	36	0.2221	0.3685	-0.0077	-0.0192	0.0607	1.5350
2.3800	-1.4500	2.2000	37	0.2051	0.3505	-0.0078	-0.0577	0.0678	1.5302
2.3800	-0.7800	4.6000	38	0.1194	0.3586	-0.0063	-0.0165	0.0904	1.5383
2.3800	-0.7800	2.2000	39	0.1101	0.3510	-0.0085	-0.0345	0.0151	1.5432
2.3800	-0.7800	1.4500	40	0.1055	0.3489	-0.0088	-0.0523	0.0516	1.5380
2.4200	-0.8200	2.2000	41	0.1081	0.3565	-0.0107	-0.0225	0.0809	0.0000
2.7000	0.0000	4.6000	42	-0.0010	0.4170	-0.0151	-0.0619	0.1414	1.5449
2.7000	0.0000	2.2000	43	-0.0149	0.4025	-0.0142	0.0131	0.1918	1.5556
2.7000	0.0000	1.4500	44	-0.0148	0.3983	-0.0137	-0.0606	-0.0149	1.5426
2.3800	0.7800	4.6000	45	-0.1219	0.3678	-0.0175	-0.1019	0.1009	1.5461
2.3800	0.7800	2.2000	46	-0.1314	0.3545	-0.0174	-0.0781	0.0050	1.5243
2.3800	0.7800	1.4500	47	-0.1347	0.3493	-0.0175	-0.0310	0.0434	1.5376
2.3800	1.4500	4.6000	48	-0.2254	0.3679	-0.0243	-0.0932	0.1137	0.0000
3.3000	-1.6800	4.6000	49	0.2575	0.5107	-0.0124	-0.0877	0.0710	1.5508
3.3000	-1.6300	2.2000	50	0.2400	0.4905	-0.0124	-0.0890	0.0711	1.5169
3.3000	-0.8000	4.6000	51	0.1210	0.5109	-0.0198	-0.0861	0.1405	1.5411
3.3000	-0.8000	2.2000	52	0.1069	0.4904	-0.0199	-0.0819	0.1113	1.5053
3.3000	0.0000	4.6000	53	-0.0021	0.5099	-0.0263	-0.0697	0.2057	1.5597
3.3000	0.0000	2.2000	54	-0.0127	0.4911	-0.0267	-0.0886	0.1723	1.4704
3.3000	1.6000	4.6000	55	-0.2487	0.5102	-0.0411	-0.0909	0.2008	0.0000
4.5000	-1.9500	4.6000	56	0.2969	0.6958	-0.0187	-0.1227	0.0714	1.5395
4.5000	-1.9500	3.1500	57	0.2878	0.6803	-0.0187	-0.0963	0.0708	1.4837
4.6000	-1.7500	4.0000	58	0.2642	0.7343	-0.0232	-0.0945	-0.0000	1.5046
5.1000	-1.5500	4.6000	59	0.2373	0.7883	-0.0276	-0.1234	0.0641	1.5523
5.1000	-1.5500	3.6000	60	0.2319	0.7735	-0.0278	-0.1457	0.0704	1.4747
5.1000	0.0000	6.2000	61	-0.0112	0.8623	-0.0506	-0.5981	-0.0578	1.4742
5.1000	0.0000	4.6000	62	-0.0019	0.7884	-0.0506	-0.1886	-0.0578	1.4742
5.1000	0.0000	3.6000	63	0.0050	0.7730	-0.0505	-0.1378	-0.0751	1.4735
5.1000	1.9500	4.6000	64	-0.3030	0.7883	-0.0757	-0.1314	0.1436	0.0000
5.7000	-1.9500	4.6000	65	0.2990	0.8809	-0.0250	-0.1107	0.0092	0.0000
6.2800	-1.3000	4.6000	66	0.2110	0.9705	-0.0314	-0.1106	0.0051	0.0000
6.4800	1.3000	4.6000	67	-0.2150	1.0014	-0.0748	-0.1506	0.0406	0.0000
7.0500	0.0000	4.6000	68	-0.0020	1.0893	-0.0511	-0.1642	0.0096	0.0000

SCALED COORDINATES
DIVIDE BY 10.000

SCALED DISPLACEMENTS/ROTATIONS FOR LOAD CASE 2 DIVIDE BY 100.00
* INDICATES A CONSTRAINT HAS BEEN APPLIED NODES 1- 68

COORDINATES X	Y	Z	NODE NO.	DISPLACEMENTS UX	UY	UZ	ROTATIONS PHIX	PHIY	PHIZ
-0.2000	0.0000	4.6000	1	0.0023	-0.0983	0.0003	-0.0401	0.1075	0.0000
-0.2000	0.0000	2.2000	2	0.0102	-0.1032	0.0012	-0.0863	0.0633	0.0000
0.0000	-0.2000	4.6000	3	0.0967	-0.0008	0.0019	-0.0352	0.1000	6.2875
0.0000	-0.2000	2.2000	4	0.0875	-0.0231	0.0025	-0.1095	0.1666	4.5284
0.0000	0.0000	5.4000	5	*	*	*	*	*	*
0.0000	0.0000	4.6000	6	0.0023	-0.0008	-0.0002	-0.0505	-0.0322	1.1148
0.0000	0.0000	2.2000	7	0.0101	-0.0225	0.0004	0.0351	0.0282	2.5245
0.0000	0.0000	0.7500	8	*	*	*	0.2154	0.0900	2.5245
0.0000	0.2000	4.6000	9	-0.0985	-0.0005	-0.0013	-0.1662	0.2218	0.0000
0.0000	0.2000	2.2000	10	-0.0738	-0.0221	0.0009	0.0229	0.0459	0.0000
0.6700	-0.9200	4.6000	11	0.4586	0.3352	0.0077	-0.2886	0.1291	4.6175
0.6700	-0.9200	2.2000	12	0.4058	0.2695	0.0063	-0.4438	0.2253	4.4872
0.8200	-0.7800	4.6000	13	0.3864	0.4098	0.0026	-0.2929	0.0475	4.9891
0.8200	-0.7800	2.2000	14	0.3551	0.3348	0.0013	-0.1929	0.1370	4.7369
0.8200	-0.7800	1.4500	15	0.3526	0.3249	-0.0007	-0.2055	0.0562	4.9070
0.7800	-0.8200	2.2000	16	0.3591	0.3219	0.0017	-0.4082	0.2060	0.0000
0.5000	0.0000	4.6000	17	-0.0002	0.2491	-0.0149	-0.2183	0.3257	4.9866
0.5000	0.0000	2.2000	18	0.0048	0.1902	-0.0153	-0.0950	0.4510	4.9788
0.5000	0.0000	1.4500	19	-0.0327	0.1679	-0.0118	-0.2913	0.3235	4.9530
0.8200	0.7800	4.6000	20	-0.3896	0.4094	-0.0374	-0.2081	0.1557	5.0136
0.8200	0.7800	2.2000	21	-0.4220	0.3628	-0.0416	-0.3932	0.0721	5.2061
0.8200	0.7800	1.4500	22	-0.4202	0.3276	-0.0446	-0.3726	0.0111	5.0044
0.6700	0.9200	4.6000	23	-0.4598	0.3344	-0.0376	-0.2284	0.2471	0.0000
1.6000	-1.3000	4.6000	24	0.6484	0.8002	-0.0003	0.0492	0.1741	5.0921
1.6000	-1.3000	2.2000	25	0.5932	0.7241	-0.0006	-0.3364	0.2048	5.0264
1.6000	-1.1000	4.6000	26	0.5489	0.8001	0.0001	-0.0202	0.1315	5.0131
1.6000	-1.1000	2.2000	27	0.5196	0.7363	0.0015	-0.2427	0.0847	5.0447
1.6000	-1.1000	1.4500	28	0.5106	0.7114	0.0020	-0.3164	0.1167	4.9820
1.6000	-1.1500	2.2000	29	0.5192	0.7244	-0.0057	-0.3508	0.1901	0.0000
1.6000	0.0000	4.6000	30	-0.0003	0.7990	-0.0202	-0.2065	-0.0287	0.0000
1.6000	0.0000	1.4500	31	-0.0343	0.7120	-0.0306	-0.2702	0.1434	0.0000
1.6000	1.1000	4.6000	32	-0.5497	0.7987	-0.0580	-0.4042	0.1139	4.9676
1.6000	1.1000	2.2000	33	-0.5739	0.7324	-0.0589	-0.2797	-0.0483	4.7290
1.6000	1.1000	1.4500	34	-0.5794	0.7126	-0.0586	-0.2055	0.1026	4.8951
1.6000	1.3000	4.6000	35	-0.6494	0.7987	-0.0660	-0.4197	0.1683	0.0000
2.3800	-1.4500	4.6000	36	0.7226	1.1896	-0.0126	-0.0679	0.1719	4.9458
2.3800	-1.4500	2.2000	37	0.6674	1.1082	-0.0131	-0.2783	0.2415	4.8345
2.3800	-0.7800	4.6000	38	0.3902	1.1900	-0.0164	-0.0870	0.3125	4.9717
2.3800	-0.7800	2.2000	39	0.3655	1.1110	-0.0173	-0.2083	0.0230	4.9066
2.3800	-0.7800	1.4500	40	0.3518	1.0972	-0.0184	-0.2596	0.1394	4.9344
2.4200	-0.8200	2.2000	41	0.3590	1.1275	-0.0277	-0.1186	0.3060	0.0000
2.7000	0.0000	4.6000	42	0.0009	1.3464	-0.0441	-0.2978	0.5521	4.9994
2.7000	0.0000	2.2000	43	-0.0319	1.2744	-0.0419	0.0905	0.7946	5.0461
2.7000	0.0000	1.4500	44	-0.0349	1.2558	-0.0409	-0.2809	-0.0506	4.9705
2.3800	0.7800	4.6000	45	-0.3903	1.1873	-0.0610	-0.5232	0.4065	5.0037
2.3800	0.7800	2.2000	46	-0.4160	1.1169	-0.0588	-0.3043	-0.0508	4.9454
2.3800	0.7800	1.4500	47	-0.4212	1.0981	-0.0585	-0.2079	0.0819	4.9530
2.3800	1.4500	4.6000	48	-0.7251	1.1676	-0.0953	-0.4706	0.4937	0.0000
3.3000	-1.6800	4.6000	49	0.8372	1.6497	-0.0257	-0.4412	0.2739	5.0951
3.3000	-1.6800	2.2000	50	0.7784	1.5533	-0.0256	-0.4246	0.2097	4.9376
3.3000	-0.8000	4.6000	51	0.3957	1.6504	-0.0613	-0.3780	0.6421	4.9905
3.3000	-0.8000	2.2000	52	0.3552	1.5527	-0.0611	-0.4843	0.3114	4.6837
3.3000	0.0000	4.6000	53	-0.0028	1.6472	-0.0926	-0.3586	0.9396	5.0923
3.3000	0.0000	2.2000	54	-0.0245	1.5551	-0.0934	-0.3903	0.7426	4.6679
3.3000	1.6000	4.6000	55	-0.8007	1.6481	-0.1725	-0.4901	0.9621	0.0000
4.5000	-1.9200	4.6000	56	0.9713	2.2485	-0.0432	-1.0273	0.0021	4.2453
4.5000	-1.9200	3.1500	57	0.9315	2.1683	-0.0434	0.4006	1.0492	4.6513
4.8000	-1.7500	4.0000	58	0.8826	2.3233	-0.0624	-0.6035	-0.0000	4.9464
5.1000	-1.5500	4.6000	59	0.7720	2.5478	-0.0814	-0.7503	0.1210	6.0092
5.1000	-1.5500	3.6000	60	0.7578	2.4703	-0.0823	-0.2093	0.4622	5.5158
5.1000	0.0000	6.2000	61	-0.0823	2.9490	-0.2002	-3.2593	-0.5016	4.0249
5.1000	0.0000	4.6000	62	-0.0021	2.5487	-0.2002	-0.9870	-0.5016	4.0249
5.1000	0.0000	3.6000	63	0.0549	2.4685	-0.2001	-0.7107	-0.6017	4.0199
5.1000	1.9500	4.6000	64	-0.9766	2.5480	-0.3298	-0.6818	0.6228	0.0000
5.7000	-1.9500	4.6000	65	0.9719	2.8476	-0.0527	-0.6345	-0.0784	0.0000
6.2800	-1.3800	4.6000	66	0.6870	3.1377	-0.0760	-0.5841	-0.1810	0.0000
6.4800	1.3800	4.6000	67	-0.6915	3.2378	-0.3026	-0.8227	0.0413	0.0000
7.0500	0.0000	4.6000	68	-0.0023	3.5223	-0.1679	-0.8679	-0.1123	0.0000

SCALED COORDINATES — DIVIDE BY 10.000

SCALED DISPLACEMENTS/ROTATIONS FOR LOAD CASE 3 — DIVIDE BY 100.00 — NODES 1 THRU 68

* INDICATES A CONSTRAINT HAS BEEN APPLIED

COORDINATES X	Y	Z	NODE NO.	DISPLACEMENTS UX	UY	UZ	ROTATIONS PHIX	PHIY	PHIZ
-0.2000	0.0000	4.6000	1	-0.0010	-0.0782	0.0028	-0.0001	0.2774	0.0000
-0.2000	0.0000	2.2000	2	-0.0276	-0.0706	-0.0001	-0.0981	0.0313	0.0000
0.0000	-0.2000	4.6000	3	0.0751	0.0000	0.0010	0.0372	0.2860	5.0866
0.0000	-0.2000	2.2000	4	0.0262	-0.0143	0.0016	-0.0299	0.2238	3.1110
0.0000	0.0000	5.4000	5	*	*	*	*	*	*
0.0000	0.0000	4.6000	6	-0.0010	-0.0000	-0.0005	-0.0246	0.0838	0.8504
0.0000	0.0000	2.2000	7	-0.0277	-0.0138	0.0002	0.0200	-0.0424	1.8781
0.0000	0.0000	0.7500	8	*	*	*	0.1331	-0.2658	1.8781
0.0000	0.2000	4.6000	9	-0.0812	0.0002	-0.0006	-0.1553	0.4234	0.0000
0.0000	0.2000	2.2000	10	-0.0873	-0.0134	0.0011	0.0291	-0.0404	0.0000
0.6700	-0.9200	4.6000	11	0.3653	0.2689	-0.0068	-0.3050	0.3254	3.6853
0.6700	-0.9200	2.2000	12	0.2547	0.1956	-0.0065	-0.5407	0.4751	3.8510
0.8200	-0.7800	4.6000	13	0.3092	0.3285	-0.0150	-0.3243	0.2547	3.9595
0.8200	-0.7800	2.2000	14	0.2151	0.2417	-0.0170	-0.1278	0.4131	3.5925
0.8200	-0.7800	1.4500	15	0.1987	0.2380	-0.0204	-0.1734	0.2595	3.8770
0.7800	-0.8200	2.2000	16	0.2199	0.2357	-0.0166	-0.4758	0.4327	0.0000
0.5000	0.0000	4.6000	17	-0.0022	0.2000	-0.0276	-0.2153	0.6029	3.9837
0.5000	0.0000	2.2000	18	-0.0332	0.1378	-0.0263	0.0505	0.8483	3.9870
0.5000	0.0000	1.4500	19	-0.1076	0.1142	-0.0206	-0.3108	0.7109	3.9493
0.8200	0.7800	4.6000	20	-0.3132	0.3282	-0.0564	-0.1640	0.4229	4.0131
0.8200	0.7800	2.2000	21	-0.4063	0.2673	-0.0625	-0.4967	0.3195	4.3316
0.8200	0.7800	1.4500	22	-0.4173	0.2425	-0.0670	-0.4395	0.1977	4.0232
0.6700	0.9200	4.6000	23	-0.3693	0.2683	-0.0520	-0.1748	0.5295	0.0000
1.6000	-1.3000	4.6000	24	0.5175	0.6408	-0.0367	0.0632	0.4214	4.0766
1.6000	-1.3000	2.2000	25	0.4008	0.5512	-0.0372	-0.3844	0.4791	4.0507
1.6000	-1.1000	4.6000	26	0.4378	0.6408	-0.0361	-0.0008	0.3752	4.0126
1.6000	-1.1000	2.2000	27	0.3510	0.5736	-0.0349	-0.2924	0.3173	4.0520
1.6000	-1.1000	1.4500	28	0.3241	0.5464	-0.0340	-0.3407	0.3561	3.9838
1.6000	-1.1500	2.2000	29	0.3416	0.5519	-0.0431	-0.4127	0.4521	0.0000
1.6000	0.0000	4.6000	30	-0.0017	0.6398	-0.0569	-0.3239	0.2071	0.0000
1.6000	0.0000	1.4500	31	-0.1102	0.5475	-0.0693	-0.2852	0.3872	0.0000
1.6000	1.1000	4.6000	32	-0.4410	0.6395	-0.0973	-0.3918	0.3666	3.9683
1.6000	1.1000	2.2000	33	-0.5219	0.5678	-0.0986	-0.3182	0.1738	3.6850
1.6000	1.1000	1.4500	34	-0.5448	0.5487	-0.0979	-0.2618	0.3408	3.8792
1.6000	1.3000	4.6000	35	-0.5208	0.6395	-0.1051	-0.4145	0.4045	0.0000
2.3800	-1.4500	4.6000	36	0.5771	0.9524	-0.0687	-0.0444	0.4353	3.9639
2.3800	-1.4500	2.2000	37	0.4609	0.8642	-0.0692	-0.2970	0.4878	3.9492
2.3800	-0.7800	4.6000	38	0.3110	0.9525	-0.0716	-0.1010	0.5792	3.9777
2.3800	-0.7800	2.2000	39	0.2321	0.8624	-0.0730	-0.1558	0.2259	3.9686
2.3800	-0.7800	1.4500	40	0.1970	0.8536	-0.0752	-0.2434	0.4087	3.9423
2.4200	-0.8200	2.2000	41	0.2102	0.8601	-0.0853	-0.1198	0.5381	0.0000
2.7000	0.0000	4.6000	42	-0.0005	1.0777	-0.1104	-0.3182	0.8222	4.0010
2.7000	0.0000	2.2000	43	-0.1049	1.0002	-0.1056	0.0950	1.0573	4.0424
2.7000	0.0000	1.4500	44	-0.1117	0.9808	-0.1030	-0.2967	0.0453	3.9658
2.3800	0.7800	4.6000	45	-0.3135	0.9504	-0.1180	-0.4042	0.6036	4.0038
2.3800	0.7800	2.2000	46	-0.3931	0.8817	-0.1169	-0.3439	0.1484	3.8603
2.3800	0.7800	1.4500	47	-0.4192	0.8560	-0.1176	-0.2408	0.3401	3.9358
2.3800	1.4500	4.6000	48	-0.5813	0.9506	-0.1482	-0.4100	0.6220	0.0000
3.3000	-1.6800	4.6000	49	0.6691	1.3206	-0.1050	-0.4262	0.5123	4.0571
3.3000	-1.6800	2.2000	50	0.5504	1.2251	-0.1051	-0.4218	0.4783	3.9672
3.3000	-0.8000	4.6000	51	0.3157	1.3211	-0.1399	-0.3915	0.7975	3.9946
3.3000	-0.8000	2.2000	52	0.2072	1.2247	-0.1404	-0.4329	0.5966	3.8214
3.3000	0.0000	4.6000	53	-0.0032	1.3184	-0.1704	-0.3180	1.0020	4.0680
3.3000	0.0000	2.2000	54	-0.0999	1.2264	-0.1727	-0.4175	0.9430	3.7639
3.3000	1.6000	4.6000	55	-0.6417	1.3191	-0.2300	-0.3704	0.9721	0.0000
4.5000	-1.9500	4.6000	56	0.7766	1.8001	-0.1534	-0.7456	0.3514	3.6053
4.5000	-1.9500	3.1500	57	0.7025	1.7309	-0.1535	0.0547	0.9103	3.7977
4.8000	-1.7500	4.0000	58	0.6810	1.8687	-0.1783	-0.2640	-0.0000	3.9659
5.1000	-1.5500	4.6000	59	0.6172	2.0397	-0.2031	-0.5926	0.4221	4.5269
5.1000	-1.5500	3.6000	60	0.5734	1.9793	-0.2036	-0.2909	0.6059	4.2637
5.1000	0.0000	6.2000	61	0.1390	2.3207	-0.2970	-2.2581	1.2184	3.5057
5.1000	0.0000	4.6000	62	-0.0020	2.0402	-0.2963	-0.7432	0.2084	3.5057
5.1000	0.0000	3.6000	63	-0.0104	1.9784	-0.2958	-0.5555	0.0216	3.5031
5.1000	1.9500	4.6000	64	-0.7826	2.0395	-0.4043	-0.5528	0.7676	0.0000
5.7000	-1.9500	4.6000	65	0.7771	2.[illegible]795	-0.2027	-0.5606	0.3591	0.0000
6.2800	-1.3800	4.6000	66	0.5493	2.[illegible]116	-0.2530	-0.5301	0.3345	0.0000
6.4800	1.3800	4.6000	67	-0.5542	2.[illegible]218	-0.4409	-0.6347	0.4683	0.0000
7.0500	0.0000	4.6000	68	-0.0023	2.[illegible]195	-0.3662	-0.6735	0.3823	0.0000

32	1	-.813E+05	-.731E+07	-72.4	-.206E+07	-.357E+07	-34.7	.161E+07	-.548E+07	5.4
32	2	.597E+07	-.225E+08	-74.8	-.685E+07	-.103E+08	-18.7	.701E+07	-.247E+08	9.4
32	3	.583E+07	-.257E+08	-81.6	-.303E+07	-.644E+07	21.5	.194E+08	-.104E+08	10.7
42	1	.158E+08	.506E+07	-17.7	.740E+07	-.499E+07	-9.3	-.651E+06	-.154E+08	-3.2
42	2	.650E+08	.249E+08	-22.6	.269E+08	-.161E+08	-11.5	-.863E+07	-.598E+08	-3.0
42	3	.696E+08	.366E+08	-29.8	.256E+08	-.117E+08	-16.1	-.152E+08	-.631E+08	-6.8
44	1	.153E+08	-.143E+08	-77.7	.325E+06	-.246E+08	-79.0	-.159E+08	-.351E+08	-80.9
44	2	.714E+08	-.306E+08	-77.4	.223E+07	-.758E+08	-79.1	-.667E+08	-.129E+09	-81.8
44	3	.842E+08	-.111E+08	-77.2	.921E+07	-.614E+08	-79.2	-.655E+08	-.112E+09	-83.2
46	1	.275E+08	.903E+07	-28.2	.143E+08	-.293E+07	-40.1	.256E+07	-.104E+08	-51.7
46	2	.110E+09	.500E+08	-22.6	.474E+08	-.994E+07	-38.1	-.762E+07	-.178E+08	-51.2
46	3	.107E+09	.414E+08	-18.2	.424E+08	-.475E+07	-34.7	-.998E+07	-.631E+08	-55.6
48	1	.542E+07	.277E+07	-85.1	-.383E+05	-.731E+07	80.6	-.530E+07	-.176E+08	77.7
48	2	.315E+08	.250E+08	-42.8	-.262E+06	-.236E+08	82.9	-.276E+08	-.767E+08	79.3
48	3	.301E+08	.172E+08	-37.9	-.456E+06	-.157E+09	87.6	-.214E+08	-.501E+08	77.9
63	1	.105E+08	.119E+07	-22.7	.247E+07	-.349E+07	30.1	.162E+07	-.154E+08	46.0
63	2	.540E+08	.391E+07	-24.7	.872E+07	-.137E+08	31.8	.568E+07	-.736E+08	49.5

SUBROUTINE R91320 TRIANGULAR HYBRID ELEMENT STRESSING ROUTINE
SIGMA1 IS THE MAXIMUM VALUE OF STRESS IN THE PLANE, SIGMA2 IS THE MINIMUM IN THE PLANE
THE ANGLE IS THAT OF SIGMA1 MEASURED +VE TO THE ELEMENT Y-AXIS FROM THE ELEMENT X-AXIS

ELEM. NUMB.	LOAD CASE	***POSITIVE*** SIGMA 1	SURFACE*** SIGMA 2	ANGLE	***NEUTRAL*** SIGMA 1	SURFACE*** SIGMA 2	ANGLE	***NEGATIVE*** SIGMA 1	SURFACE*** SIGMA 2	ANGLE
63	3	.395E+08	.413E+07	-31.7	.742E+07	-.101E+08	30.0	.648E+07	-.555E+08	44.2
72	1	-.279E+07	-.357E+07	78.1	.208E+07	-.318E+07	8.5	.765E+07	-.347E+07	7.2
72	2	.327E+06	-.477E+08	-26.6	.768E+07	-.112E+08	8.4	.451E+08	-.442E+07	40.7
72	3	.396E+07	-.267E+08	-18.6	.631E+07	-.743E+07	8.8	.237E+08	-.323E+07	43.2
73	1	.173E+07	.784E+06	31.9	.245E+07	-.345E+06	-27.2	.390E+07	-.220E+07	-31.1
73	2	-.203E+08	-.167E+09	-82.5	.162E+08	-.160E+07	-41.4	.181E+09	.411E+08	0.2
73	3	-.144E+08	-.854E+08	-82.7	.954E+07	-.129E+07	-36.4	.976E+08	.239E+08	0.8
74	1	-.707E+06	-.566E+07	1.0	-.150E+07	-.603E+07	28.8	-.615E+06	-.699E+07	45.4
74	2	-.855E+08	-.129E+09	73.7	-.129E+07	-.120E+08	51.5	.112E+09	.811E+08	2.2
74	3	-.484E+08	-.672E+08	72.2	-.188E+05	-.706E+07	38.7	.600E+08	.415E+08	4.3
75	1	.360E+07	-.589E+07	-50.3	.213E+07	-.531E+07	-28.3	.316E+07	-.724E+07	-8.7
75	2	-.649E+08	-.125E+09	-53.3	.148E+ 8	-.185E+08	-21.4	.132E+09	.710E+08	-3.1
75	3	-.426E+08	-.679E+08	-53.7	.932E+ 7	-.106E+08	-25.9	.706E+08	.376E+08	-6.5
76	1	-.377E+06	-.127E+08	20.3	.116E+06	-.130E+08	33.1	.102E+07	-.145E+08	43.7
76	2	-.101E+09	-.155E+09	45.1	.385E+07	-.375E+08	39.1	.111E+09	.760E+08	29.3
76	3	-.529E+06	-.865E+08	39.6	.308E+07	-.248E+08	35.0	.596E+08	.304E+08	28.4
80	1	.925E+06	-.554E+07	55.9	.192E+07	.163E+06	35.3	.662E+07	.215E+07	-18.6
80	2	.410E+07	-.327E+08	72.0	.692E+07	.121E+07	32.0	.407E+08	.414E+07	-9.1
80	3	.355E+07	-.750E+07	71.9	.519E+07	.168E+07	18.6	.163E+08	.183E+07	-5.0
82	1	.222E+07	.106E+07	-20.7	.368E+06	-.669E+06	-61.3	-.830E+06	-.305E+07	-76.9
82	2	.129E+08	.688E+07	-12.8	.167E+07	-.304E+07	-64.9	-.509E+07	-.174E+08	-78.9
82	3	.468E+07	.123E+07	-34.1	.254E+07	-.354E+07	-65.7	.156E+07	-.948E+07	-73.6
83	1	-.349E+06	-.102E+08	-2.2	.444E+06	-.189E+07	-48.2	.940E+07	-.169E+07	-79.8
83	2	-.506E+06	-.553E+08	0.3	.189E+07	-.696E+07	-50.3	.532E+08	-.756E+07	-81.4
83	3	-.316E+07	-.298E+08	-4.0	.254E+07	-.643E+07	-54.8	.301E+08	-.469E+07	-78.9

THIN FACET SHELL STRESSES/MOMENTS ELEMENT.TYPE=44200, 4 NODES

ELEM NO	LOAD CASE	NODE NO	PRINCIPAL-BEND-MOMENTS MOST.PLUS	MOST.MINUS	PRINCIPAL-STRESSES MOST.PLUS	MOST.MINUS	THICKNESS POSITION	ANGLES OF MOST POSITIVE VALUE TO X AXES ELEMENT	GLOBAL
								10.4	1090.0
6	1	3	-138.	-598.				-81.0	1090.0
					.993E+08	.219E+08	TOP	60.3	1090.0
					.125E+07	-.264E+07	MIDDLE	11.9	1090.0
					-.240E+08	-.100E+09	BOTTOM	-89.6	1090.0
								0.5	1090.0
6	1	6	.718E+04	.216E+04				55.5	1090.0
					-.361E+09	-.120E+10	TOP	-89.8	1090.0
					.192E+07	-.271E+07	MIDDLE	41.9	1090.0
					.120E+10	.359E+09	BOTTOM	-62.2	1090.0
								76.2	1090.0
8	1	4	46.1	-149.				52.2	1090.0
					.341E+08	.355E+07	TOP	-89.4	1090.0
					.180E+08	.248E+07	MIDDLE	1.4	1090.0
					.220E+08	-.188E+08	BOTTOM	78.3	1090.0
								89.8	1090.0
8	1	7	.267E+04	798.					
					-.129E+09	-.428E+09	TOP		
					.184E+08	.267E+07	MIDDLE		
					.463E+09	.137E+09	BOTTOM		
								11.9	1090.0
8	2	3	-422.	-.197E+04				-80.1	1090.0
					.321E+09	.669E+08	TOP	51.1	1090.0
					.406E+07	-.138E+08	MIDDLE	13.8	1090.0
					-.731E+08	-.334E+09	BOTTOM	-89.7	1090.0
8	2	6	.233E+05	.701E+04					

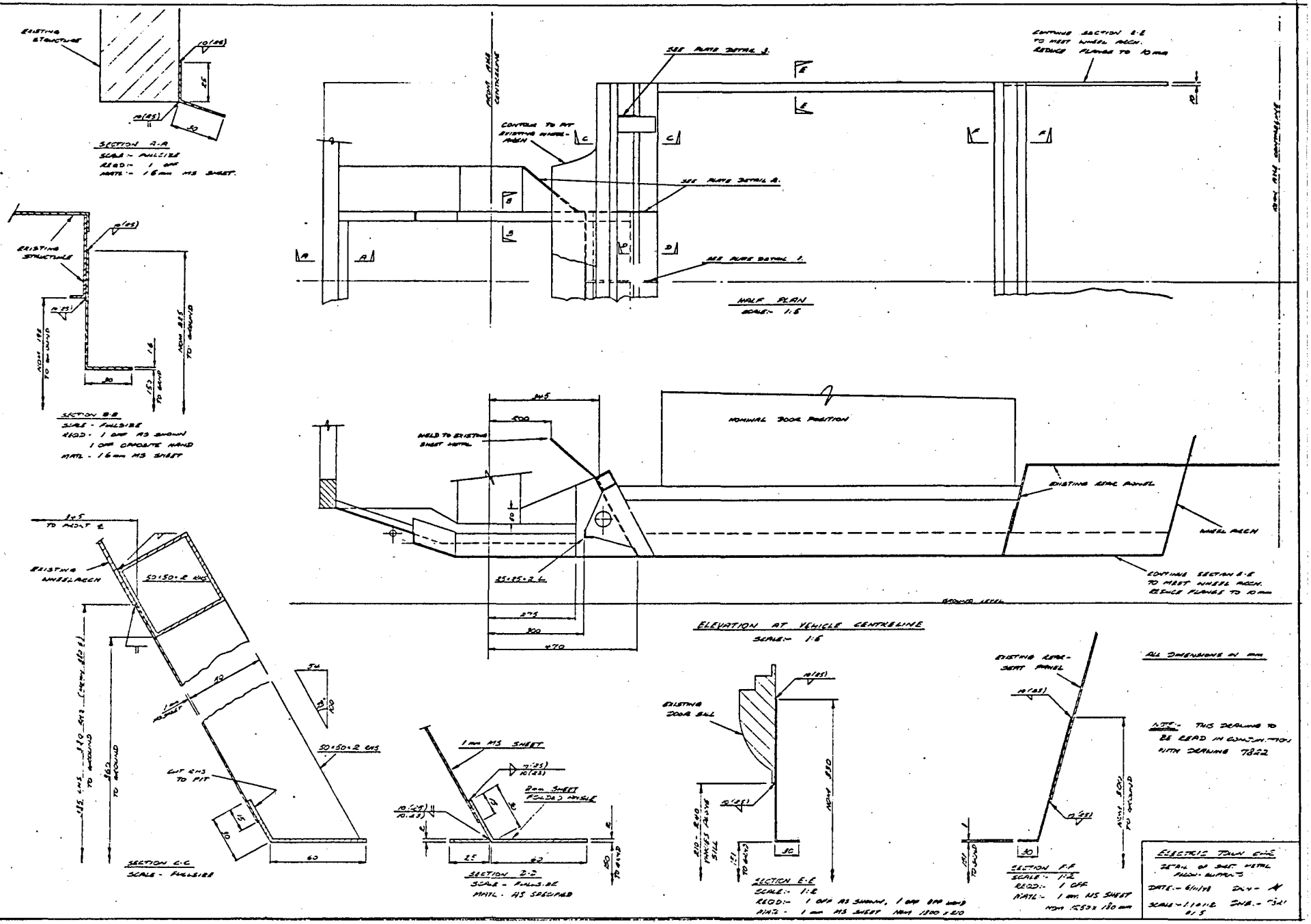
						.147E+08	-.546E+08	MIDDLE	38.6	1090.0
						.220E+08	-.662E+08	BOTTOM	39.1	1090.0
77	2	60	189.	-384.					12.4	1090.0
						.688E+08	-.289E+08	TOP	-80.0	1090.0
						.171E+08	-.240E+08	MIDDLE	18.2	1090.0
						.622E+08	-.116E+09	BOTTOM	13.8	1090.0
77	3	62	-126.	-170.					-20.8	1090.0
						.528E+08	.139E+08	TOP	61.1	1090.0
						.122E+08	-.166E+08	MIDDLE	58.1	1090.0
						-.281E+08	-.475E+08	BOTTOM	52.1	1090.0
77	3	59	-109.	-207.					18.2	1090.0
						.603E+08	.427E+08	TOP	-72.7	1090.0
						.165E+08	.107E+08	MIDDLE	21.1	1090.0
						-.977E+07	-.389E+08	BOTTOM	18.8	1090.0
77	3	63	20.2	-30.8					41.1	1090.0
						-.176E+07	-.265E+08	TOP	45.2	1090.0
						.306E+07	-.338E+08	MIDDLE	43.9	1090.0
						.790E+07	-.412E+08	BOTTOM	43.2	1090.0
77	3	60	103.	-197.					13.7	1090.0
						.361E+08	-.127E+08	TOP	-79.0	1090.0
						.123E+08	-.114E+08	MIDDLE	19.4	1090.0
						.367E+08	-.584E+08	BOTTOM	15.1	1090.0
81	1	59	323.	-68.7					24.4	32.6
						.548E+07	-.181E+08	TOP	-68.3	-60.1
						.242E+07	.186E+06	MIDDLE	69.5	77.7
						.207E+08	-.287E+07	BOTTOM	27.1	35.3
81	1	62	-161.	-313.					47.9	56.1
						.200E+08	.109E+08	TOP	-48.9	-40.7
						.233E+07	.185E+06	MIDDLE	89.2	-82.6
						-.816E+07	-.178E+08	BOTTOM	54.3	62.5
81	1	66	-104.	-194.					33.2	41.4
						.102E+08	.603E+07	TOP	-56.1	-47.9
						-.191E+06	-.140E+07	MIDDLE	30.8	39.0
						-.642E+07	-.130E+08	BOTTOM	32.6	41.0
81	1	68	83.7	-32.9					-23.7	-15.5
						.228E+07	-.516E+07	TOP	69.9	78.1
						.587E+06	-.416E+06	MIDDLE	-80.3	-72.1
						.494E+07	-.172E+07	BOTTOM	-27.6	-19.4
81	2	59	.198E+04	-43.1					31.1	39.3
						.847E+07	-.109E+09	TOP	-61.7	-53.5
						.140E+08	.161E+07	MIDDLE	65.6	73.8
						.129E+09	.274E+07	BOTTOM	33.7	41.9
81	2	62	-.110E+04	-.184E+04					41.3	49.5
						.118E+09	.719E+08	TOP	-55.9	-47.7
						.129E+08	.124E+07	MIDDLE	88.0	-83.8
						-.584E+08	-.104E+09	BOTTOM	48.8	57.0
81	2	66	-545.	-.123E+04					26.3	34.5
						.644E+08	.317E+08	TOP	-62.9	-54.7
						-.922E+06	-.923E+07	MIDDLE	23.3	31.5
						-.336E+08	-.828E+08	BOTTOM	25.8	34.0
81	2	68	430.	-170.					-26.1	-17.9
						.106E+08	-.267E+08	TOP	67.5	75.7
						.215E+07	-.260E+07	MIDDLE	-76.9	-68.7
						.253E+08	-.101E+08	BOTTOM	-29.9	-21.7
81	3	59	790.	62.4					35.6	43.8
						-.154E+07	-.394E+08	TOP	-58.2	-50.1
						.913E+07	.112E+07	MIDDLE	55.3	63.5
						.558E+08	.564E+07	BOTTOM	38.5	46.7
81	3	62	-630.	-958.					28.4	36.6
						.608E+08	.399E+08	TOP	-72.7	-64.5
						.664E+07	-.126E+07	MIDDLE	72.1	80.3
						-.342E+08	-.558E+08	BOTTOM	39.2	47.4
81	3	66	-320.	-699.					20.5	28.7
						.405E+08	.213E+08	TOP	-70.2	-62.0
						.206E+07	-.149E+07	MIDDLE	23.9	32.1
						-.172E+08	-.434E+08	BOTTOM	20.9	29.1
81	3	68	196.	-24.9					-23.3	-15.1
						.316E+07	-.140E+08	TOP	69.7	77.9
						.182E+07	-.242E+07	MIDDLE	79.2	87.4

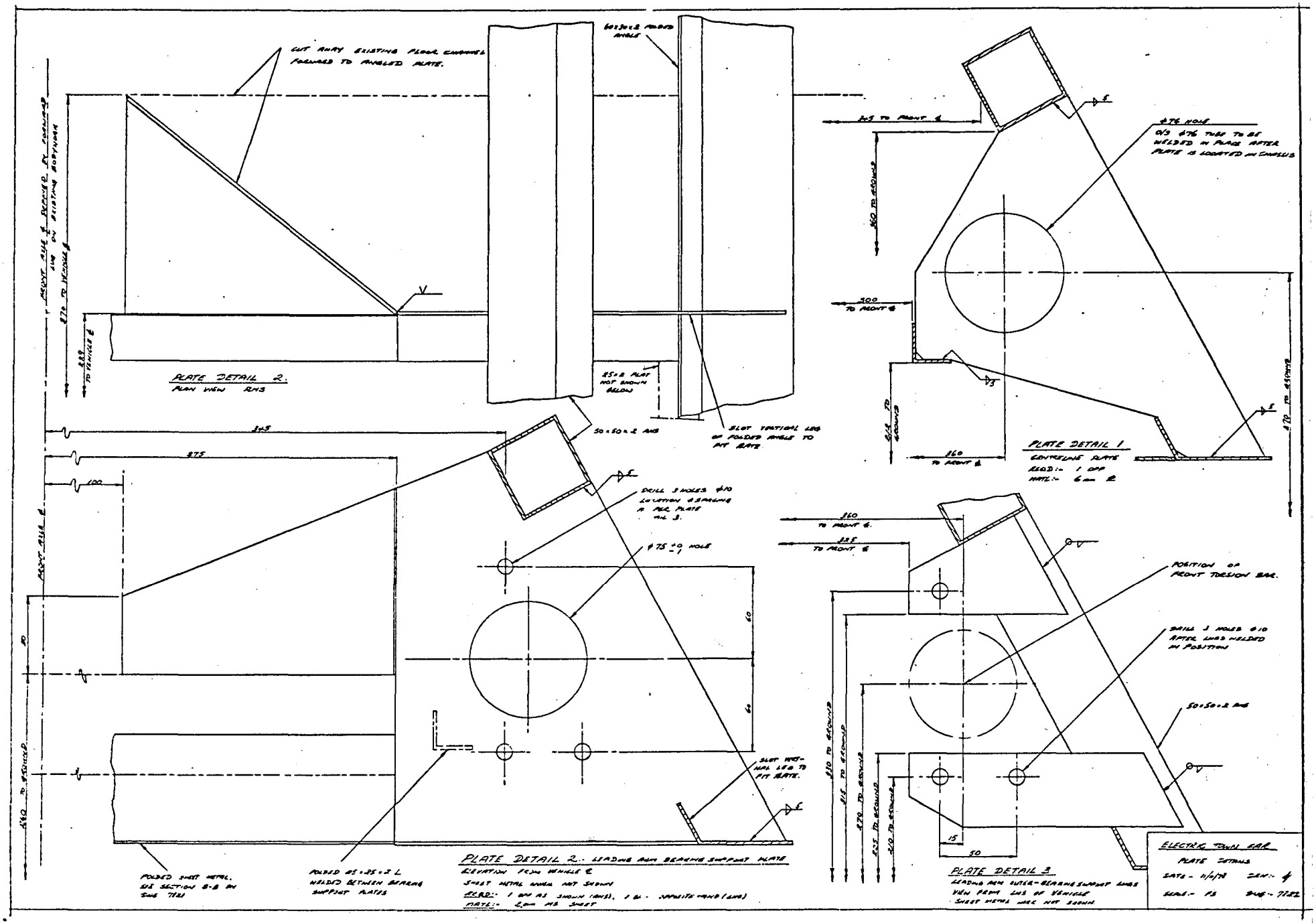
APPENDIX 6.1

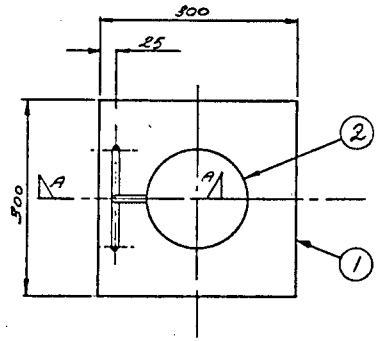
ENGINEERING WORKSHOP DRAWINGS

FOR THE MANUFACTURE OF

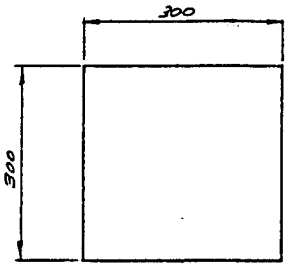
AN ELECTRIC TOWN CAR


EXISTING STRUCTURE
SECTION A-A
SCALE - FULL SIZE
REQD:- 1 OFF
MATL:- 16mm MS SHEET
SEE PLATE DETAIL 3.
SEE PLATE DETAIL 2.
SEE PLATE DETAIL 1.
CONTINUE SECTION E-E TO MEET WHEEL ARCH. REDUCE FLANGE TO 10mm
CONTOUR TO FIT EXISTING WHEEL-ARCH
HALF PLAN
SCALE:- 1:6
SECTION B-B
SCALE - FULL SIZE
REQD - 1 OFF AS SHOWN
1 OFF OPPOSITE HAND
MATL - 16mm MS SHEET
NOMINAL DOOR POSITION
WELD TO EXISTING SHEET METAL
EXISTING REAR PANEL
WHEEL ARCH
CONTINUE SECTION E-E TO MEET WHEEL ARCH. REDUCE FLANGE TO 10mm
ELEVATION AT VEHICLE CENTRELINE
SCALE:- 1:6
EXISTING WHEELARCH
SECTION C-C
SCALE - FULL SIZE
1mm MS SHEET
SECTION D-D
SCALE - FULL SIZE
MATL - MS SPECIFIED
EXISTING DOOR SILL
SECTION E-E
SCALE:- 1:2
REQD:- 1 OFF AS SHOWN, 1 OFF OPP HAND
MATL:- 1mm MS SHEET NOM 1500 x 810
EXISTING REAR-SEAT PANEL
SECTION F-F
SCALE:- 1:2
REQD:- 1 OFF
MATL:- 1mm MS SHEET NOM 1650 x 180mm
ALL DIMENSIONS IN mm
NOTE:- THIS DRAWING TO BE READ IN CONJUNCTION WITH DRAWING 7822
ELECTRIC TOWN CHIC
DETAIL OF SHEET METAL MODIFICATIONS
DATE:- 6/11/98
DWG:- A
SCALE - 1:6 1:2 1:5
SHR:- "32"

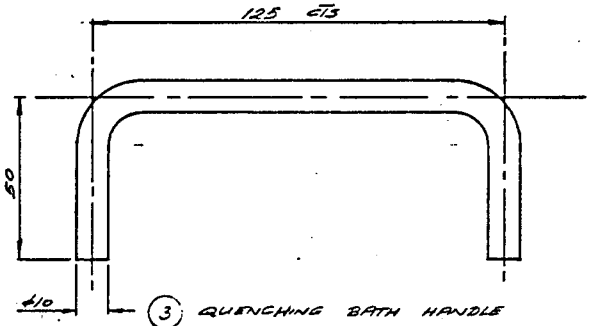

CUT AWAY EXISTING FLOOR CHANNEL FORWARD TO ANGLED PLATE.
PLATE DETAIL 2.
PLAN VIEW RHS
SLOT VERTICAL LEG OF FOLDED ANGLE TO FIT PLATE
PLATE DETAIL 1
CENTRELINE PLATE
REQD:- 1 OFF
MATL:- 6mm ℄
Ø76 HOLE
Ø/3 Ø76 THRU TO BE WELDED IN PLACE AFTER PLATE IS LOCATED ON CHASSIS
DRILL 3 HOLES Ø10 LOCATION & SPACING FROM PIVOT PLATE AS No 3.
Ø75 +0 -1 HOLE
SLOT VERTICAL LEG TO FIT PLATE.
FOLDED SHEET METAL. SEE SECTION B-B ON DWG 7182
FOLDED 65×35×3 L WELDED BETWEEN BEARING SUPPORT PLATES
PLATE DETAIL 2.. LEADING ARM BEARING SUPPORT PLATE
ELEVATION FROM WHEEL ℄
SHEET METAL WORK NOT SHOWN
REQD:- 1 OFF AS SHOWN (RHS), 1 OFF - OPPOSITE HAND (LHS)
MATL:- 2mm MS SHEET
POSITION OF FRONT TORSION BAR.
DRILL 3 HOLES Ø10 AFTER LUGS WELDED IN POSITION
50×50×2 RHS
PLATE DETAIL 3
LEADING ARM OUTER-BEARING SUPPORT LUGS
VIEW FROM LHS OF VEHICLE
SHEET METAL WORK NOT SHOWN
ELECTRIC TOWN CAR
PLATE DETAILS
DATE:- 11/6/78
SCALE:- FS
DWG:- 7182



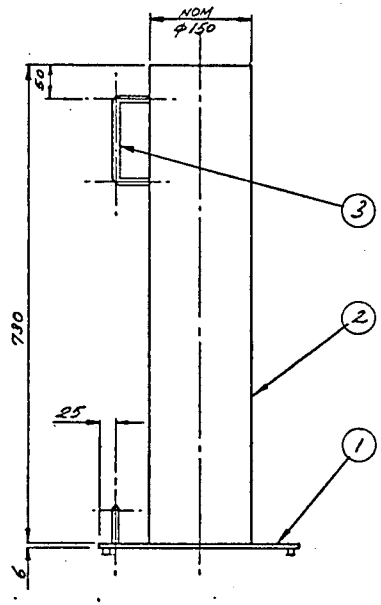
GENERAL ARRANGEMENT
PLAN VIEW



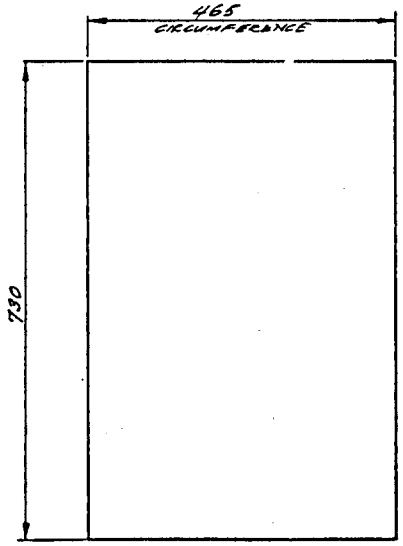
(1) BASE PLATE
REQD:- 1 OFF
MATL:- 6 mm Pl



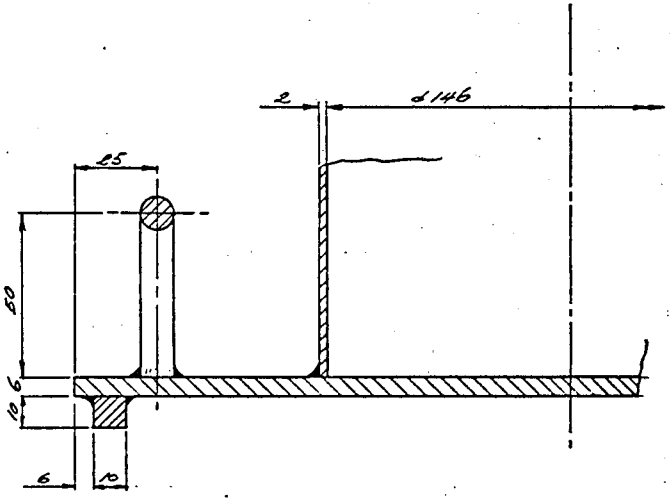
(3) QUENCHING BATH HANDLE
REQD:- 2 OFF
MATL:- ϕ 10 MS BLACK.



GENERAL ARRANGEMENT
ELEVATION.

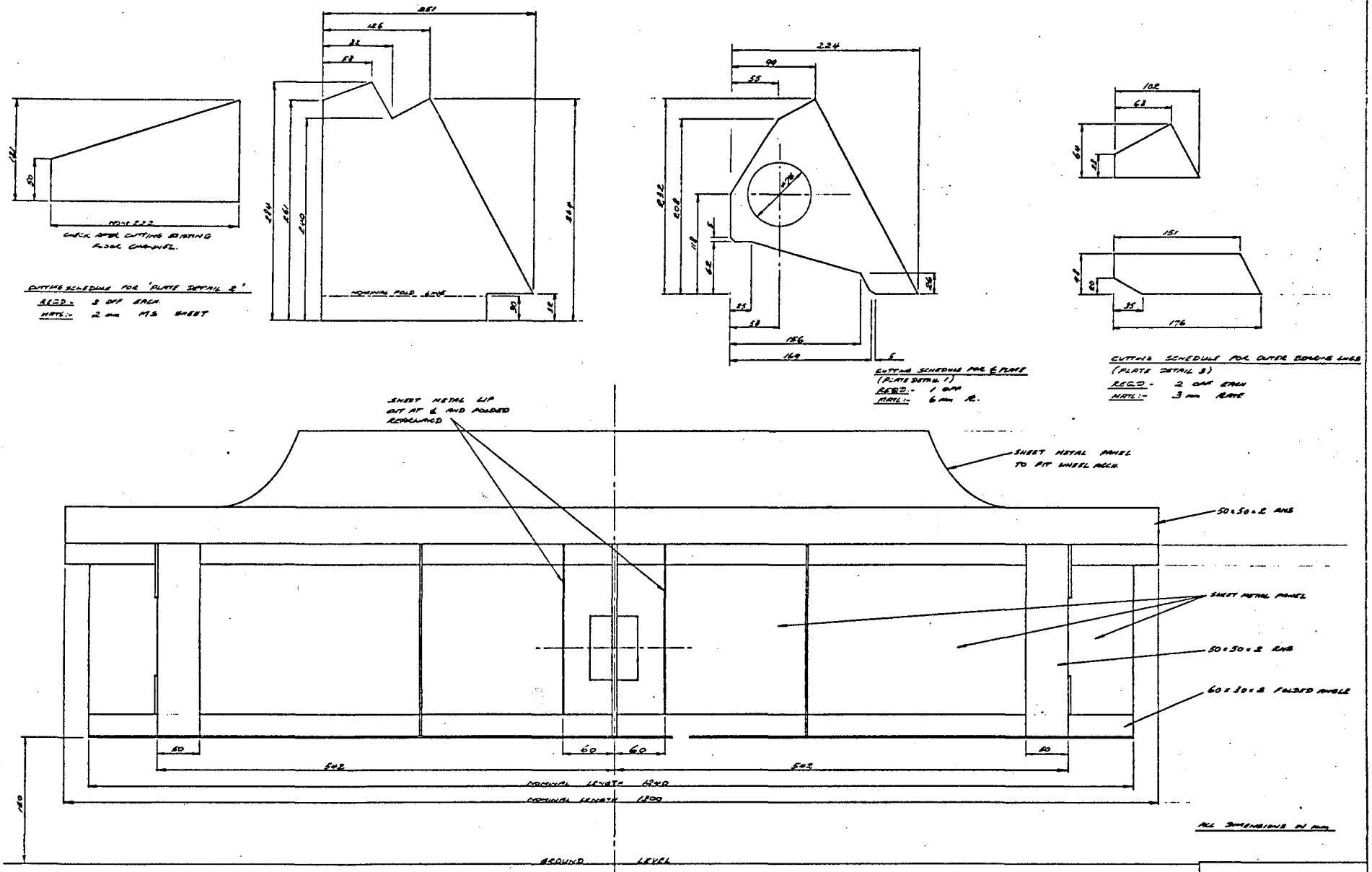


(2) CYLINDER DEVELOPMENT.
REQD:- 1 OFF
MATL:- 2 mm MS SHEET.



SECTION DETAIL A-A.

ELECTRIC TOWN CAR	
TORSION BAR QUENCHING BATH.	
DATE:- 26/11/68	DWG:- 4/
SCALE:- NTS	DRG:- 782



CUTTING SCHEDULE FOR 'PLATE DETAIL 2'
REQ'D:- 2 OFF EACH
MATL:- 2 mm M.S. SHEET

CHECK AFTER CUTTING EXISTING FLOOR CHANNEL.

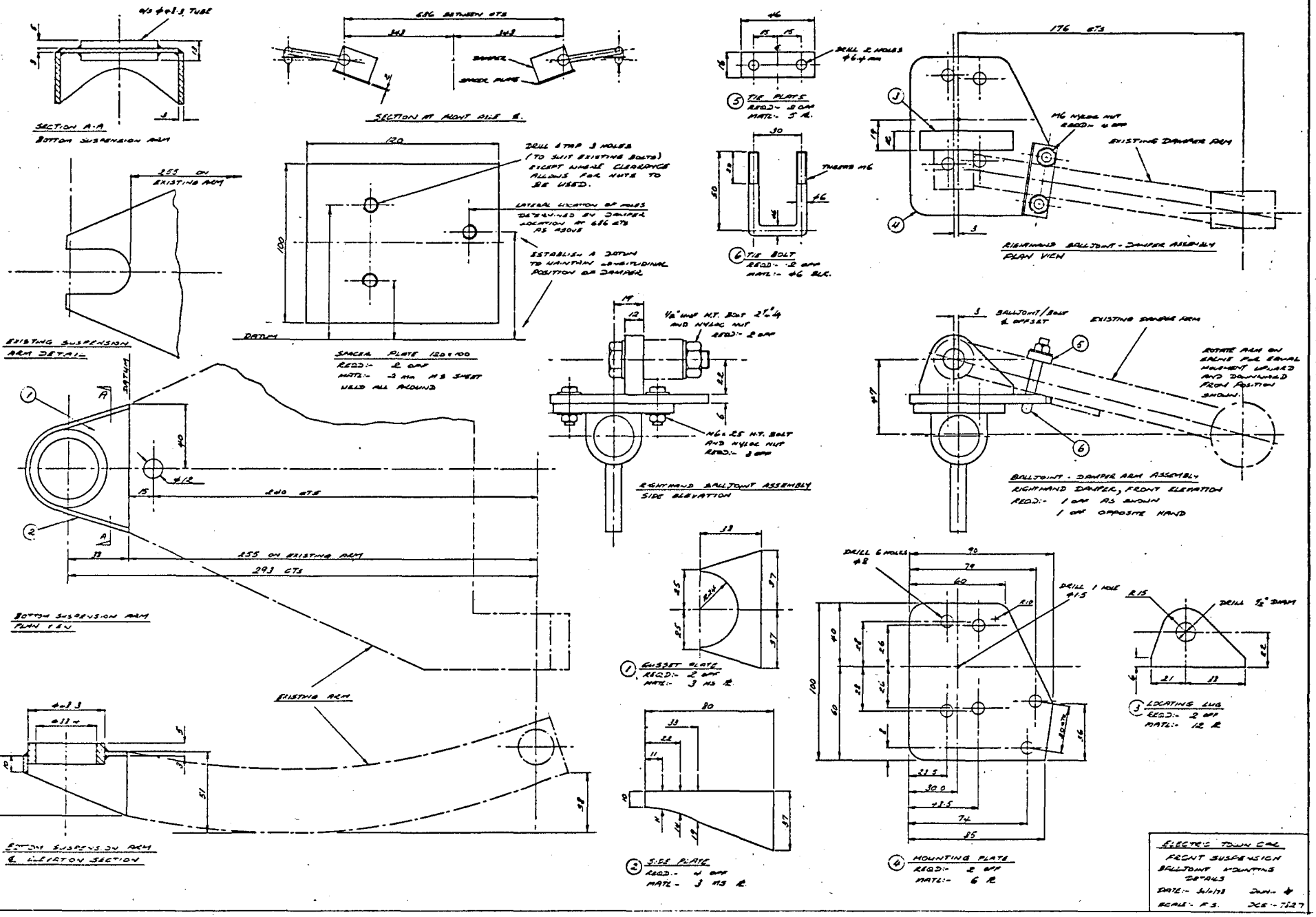
CUTTING SCHEDULE FOR 6 PLATE (PLATE DETAIL 1)
REQ'D:- 1 OFF
MATL:- 6 mm M.

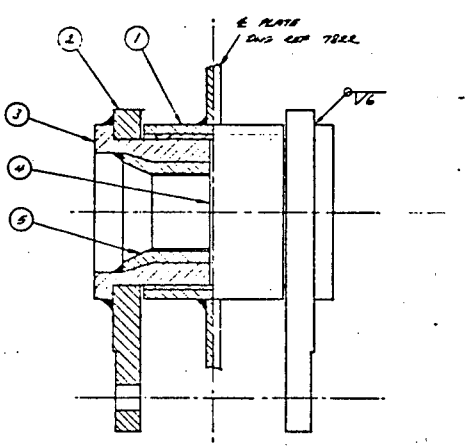
CUTTING SCHEDULE FOR OUTER BOGGING LUGS (PLATE DETAIL 3)
REQ'D - 2 OFF EACH
MATL:- 3 mm PLATE

ALL DIMENSIONS IN mm.

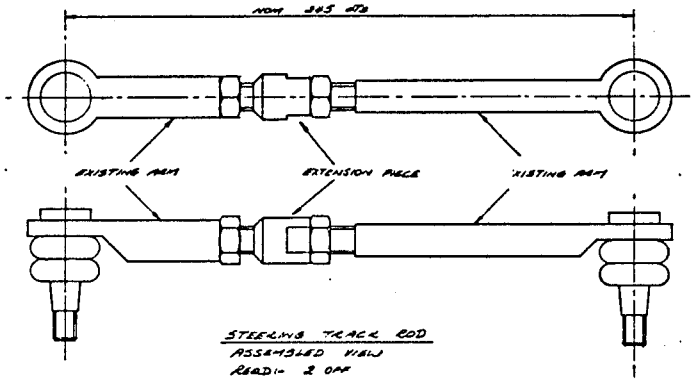
FRONT PERIMETER FRAME DETAIL
VIEW FROM REAR
SCALE:- 1:2

ELECTRIC TOWN CAR
PLATE CUTTING SCHEDULE AND PERIMETER FRAME
DATE - 12/1/78 DRN.- 4
SCALE - 1:2 DWG - 7124

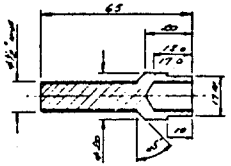

SECTION A-A
BOTTOM SUSPENSION ARM
SECTION AT FRONT AXLE ℄
SPACER
SHIM PLATE
5 TIE PLATE
REQD:- 8 OFF
MATL:- 3 ℄
6 TIE BOLT
REQD:- 2 OFF
MATL:- 40 ELC.
RIGHTHAND BALLJOINT - DAMPER ASSEMBLY
PLAN VIEW
EXISTING DAMPER ARM
EXISTING SUSPENSION ARM DETAIL
SPACER PLATE 120 x 100
REQD:- 2 OFF
MATL:- 3 mm MS SHEET
WELD ALL ROUND
RIGHTHAND BALLJOINT ASSEMBLY
SIDE ELEVATION
BALLJOINT - DAMPER ARM ASSEMBLY
RIGHTHAND DAMPER, FRONT ELEVATION
REQD:- 1 OFF AS SHOWN
1 OFF OPPOSITE HAND
ROTATE ARM ON EACH PIN EQUAL MOVEMENT UPWARD AND DOWNWARD FROM POSITION SHOWN.
BOTTOM SUSPENSION ARM
PLAN VIEW
EXISTING ARM
BOTTOM SUSPENSION ARM
& ELEVATION SECTION
1 GUSSET PLATE
REQD:- 2 OFF
MATL:- 3 MS ℄
2 SIDE PLATE
REQD:- 4 OFF
MATL- 3 MS ℄
4 MOUNTING PLATE
REQD:- 2 OFF
MATL:- 6 ℄
3 LOCATING LUG
REQD:- 2 OFF
MATL:- 12 ℄
ELECTRIC TOWN CAR
FRONT SUSPENSION
BALLJOINT MOUNTING
DETAILS
DATE:- 5/6/73
SCALE:- F.S.
DWN:-
DCE:- 7327



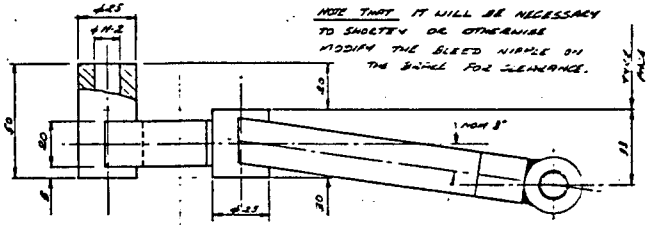
GENERAL ARRANGEMENT
FRONT PITCH ARMS BEARING ASSEMBLY

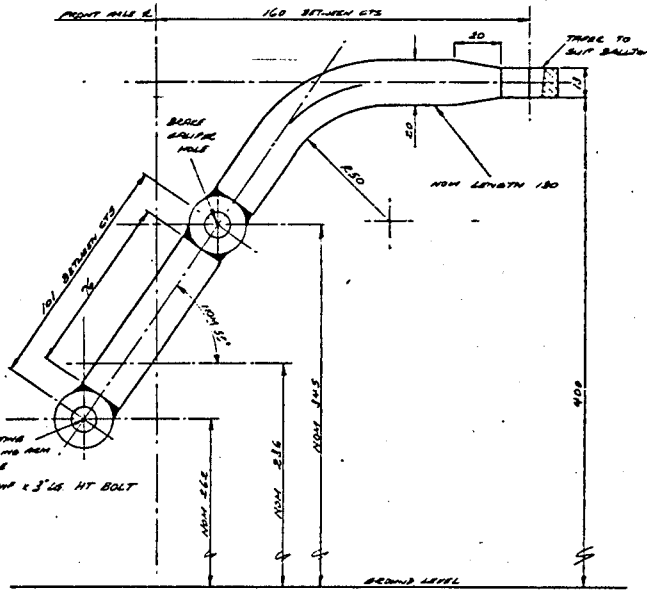


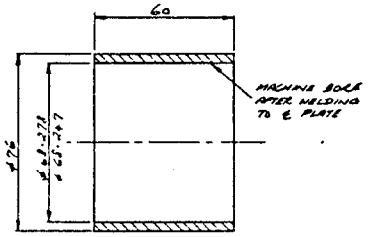
STEERING TRACK ROD
ASSEMBLED VIEW
READ:- 2 OFF



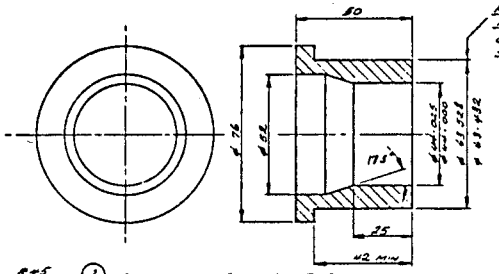
EXTENSION PIECE
READ:- 2 OFF
MATL:- ϕ20 ROUND



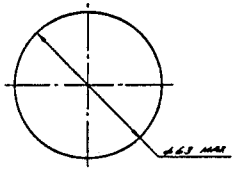




① BEARING HOUSING
READ:- 2 OFF (FRONT & REAR)
MATL:- ϕ76 STEAM TUBE

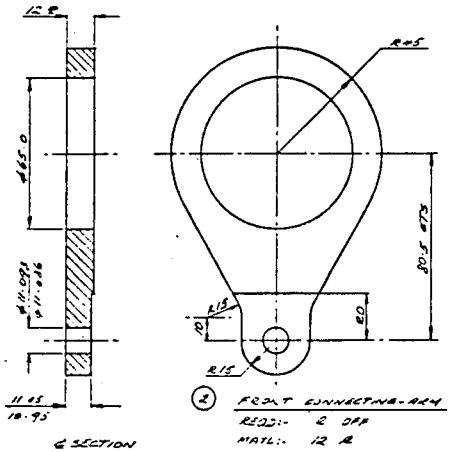


③ CONNECTING-ARM SPINDLE
READ:- 2 OFF FRONT
2 OFF REAR.
MATL:- ϕ76 ROUND.

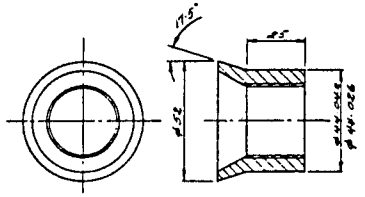


④ SEPERATING DISC
READ:- 1 OFF FRONT
1 OFF REAR
MATL:- 1/8" POLYETHYLENE

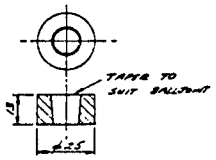
STEERING ARM
PLAN & ELEVATION.
READ:- 1 OFF AS SHOWN (RHS)
1 OFF OPPOSITE HAND (LHS)
MATL:- 20 x 20 MS BLK.
NOTE:- FABRICATE ARM IN POSITION ON SUSPENSION UPRIGHT.



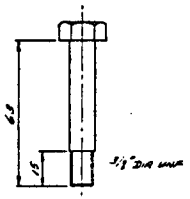
② FRONT CONNECTING-ARM
READ:- 2 OFF
MATL:- 12 ℄



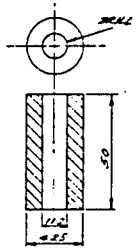
⑤ TORSION BAR SPLINE HOUSING
READ:- 2 OFF FRONT
2 OFF REAR
MATL:- EXISTING TW HOUSING



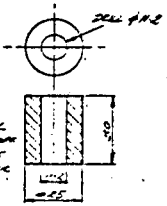
BALLJOINT BUSH
READ:- 2 OFF
MATL:- ϕ25 BAR



CALIPER BOLT
READ:- 2 OFF
MATL:- 7/16"DIA UNF HIGH TENSILE BOLT 3" LONG
(4 BOLTS REQD TOTAL)

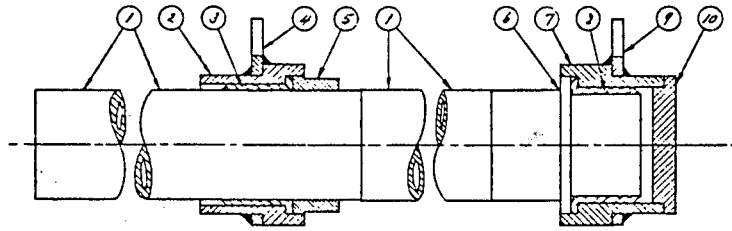


STEERING ARM BUSH
READ:- 2 OFF
MATL:- ϕ25 BAR

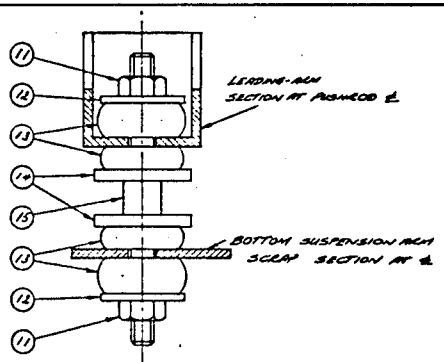


EQUIPE BUSH
READ:- 2 OFF
MATL:- ϕ25 BAR

ELECTRIC TOWN CAR
STEERING ARM CONSTRUCTIONAL DETAILS
DATE:- 31/3/71 DWG - 4
SCALE - AS DWG - 7622

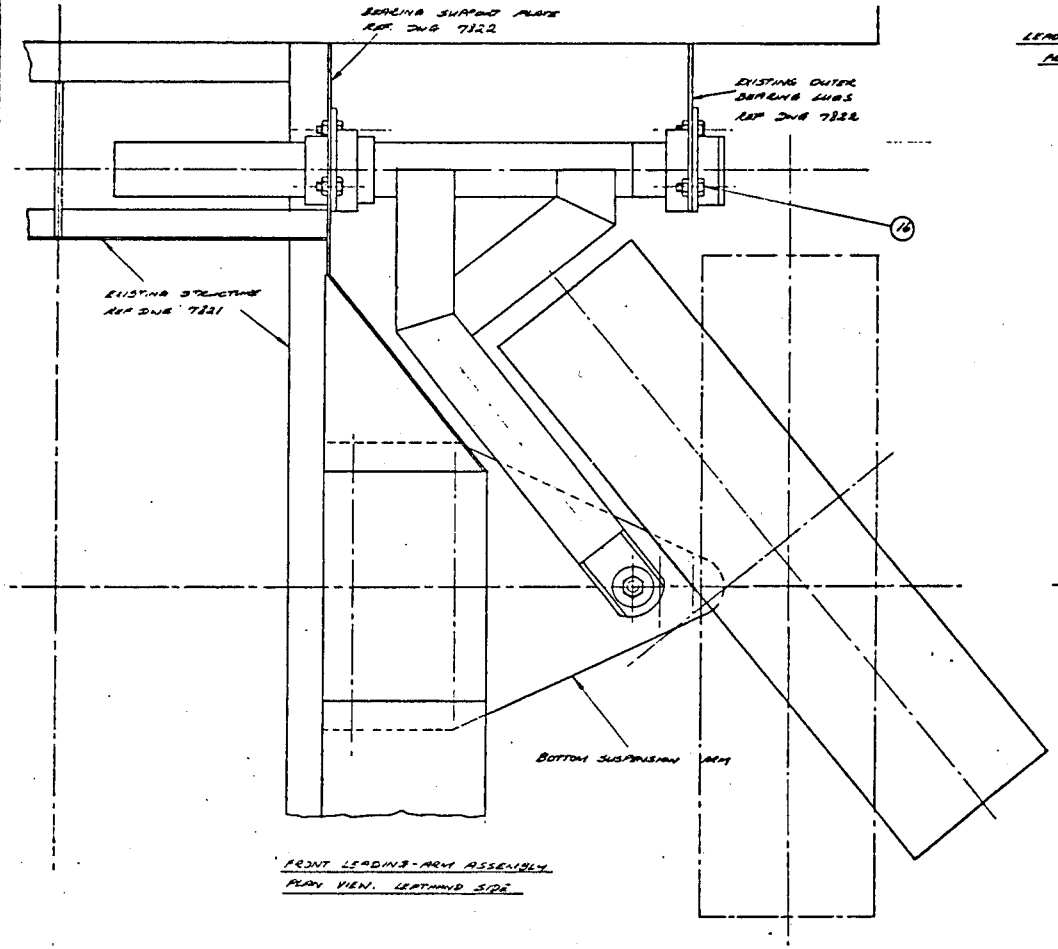


FRONT LEADING-ARM BEARING ASSEMBLY DETAIL

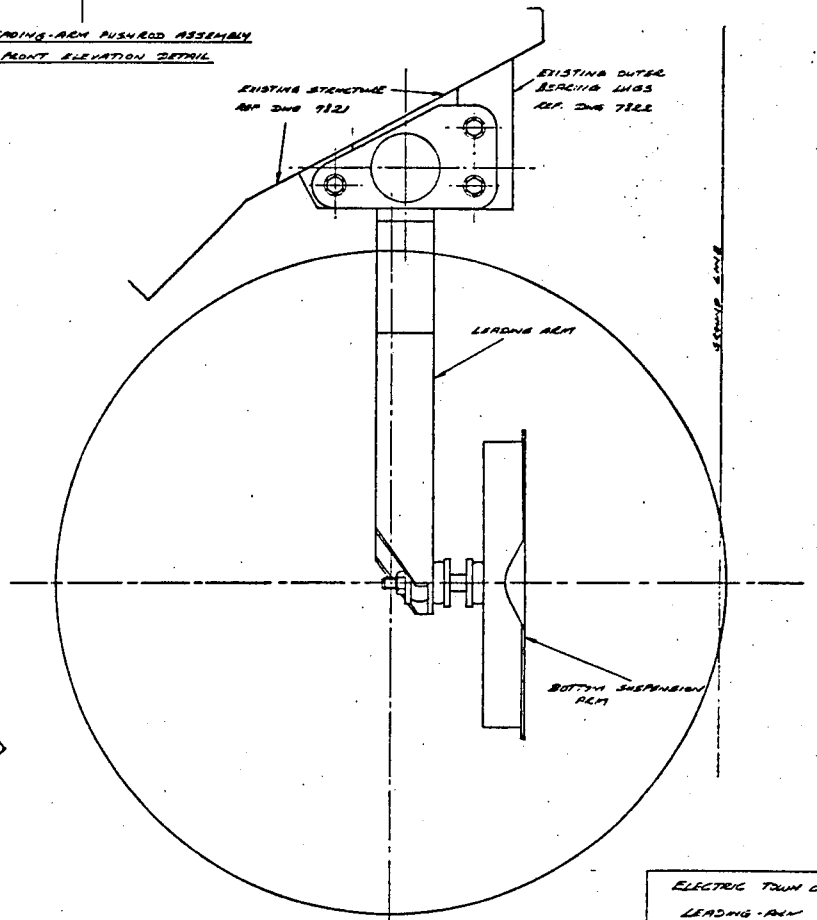


LEADING-ARM PUSHROD ASSEMBLY FRONT ELEVATION DETAIL

ITEM	DESCRIPTION	REQD	PART
1	LEADING ARM CROSS-TUBE	2	7824
2	INNER BEARING HOUSING	2	7825
3	[illegible]	-	7826
4	BEARING SUPPORT PLATE	2	7827
5	INNER BEARING SHOULDER	2	7828
6	RH TORSION BAR HOUSING	1	7829
7	OUTER BEARING HOUSING	2	7830
8	OUTER LEAF ARM	2	7831
9	BEARING SUPPORT PLATE	2	7832
10	BEARING HOUSING CAP	2	7833
11	NUT HEXG. HT	4	
12	STUD 1123-24	4	7835
13	SHOCK ABSORBER BUSH	8	
14	FLAT WASHER	4	7836
15	PUSHROD	2	7837
16	NUT & CO H.T BOLT & NUT	12	

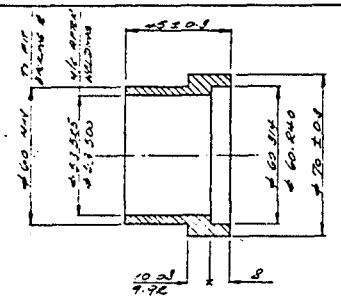


FRONT LEADING-ARM ASSEMBLY PLAN VIEW. LEFTHAND SIDE

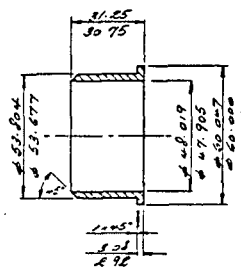


FRONT LEADING-ARM ASSEMBLY ELEVATION LEFTHAND SIDE

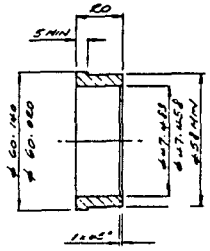
ELECTRIC TOWN CAR
LEADING-ARM ASSEMBLY DETAILS
DATE- 22/2/'58 DRN. 4.
SCALE- 1½ & 1/8 DWG - 7827



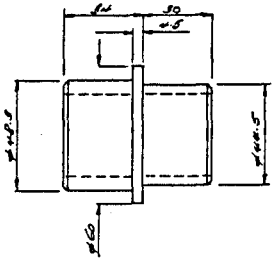
(2) INNER BEARING HOUSING
REQD:- 4 OFF
MATL:- Φ75 MS BLACK



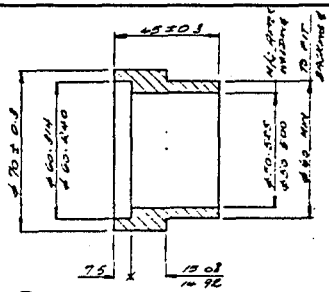
(3) INNER BEARING BUSH
REQD:- 4 OFF
MATL:- DILON PV80 BEARING MATERIAL



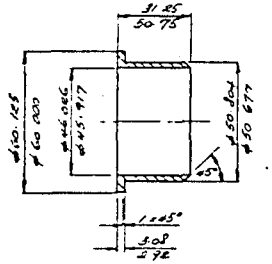
(5) INNER BEARING SHOULDER
REQD:- 4 OFF
MATL:- Φ65 MS BLACK



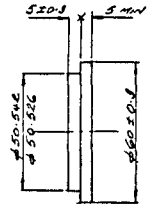
(6) EXISTING VW TORSION-BAR HOUSING
NOMINAL DIMENSIONS



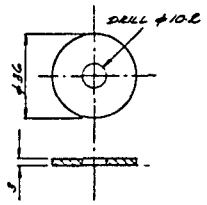
(7) OUTER BEARING HOUSING
REQD:- 4 OFF
MATL:- Φ75 BLACK



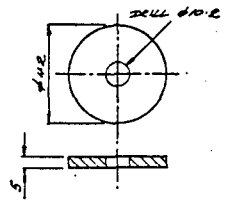
(8) OUTER BEARING BUSH
REQD:- 4 OFF
MATL:- DILON PV80 BEARING MATERIAL



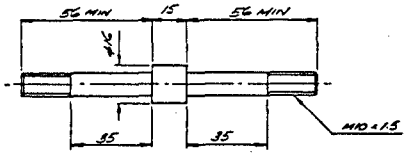
(10) BEARING HOUSING CAP
REQD:- 4 OFF
MATL:- Φ65 MS



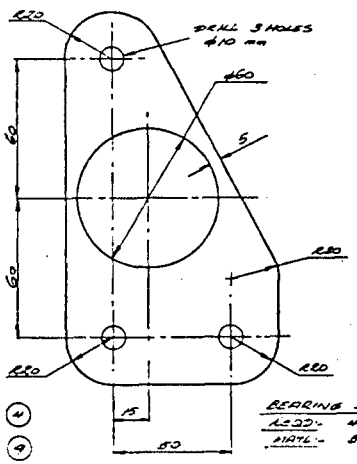
(12) 3mm WASHER
REQD:- 4 OFF



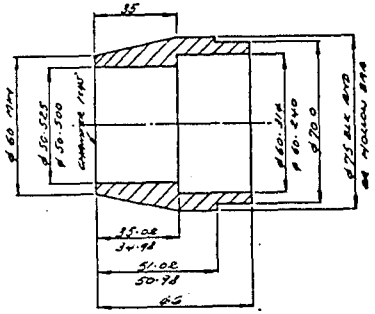
(14) 5mm WASHER
REQD:- 4 OFF



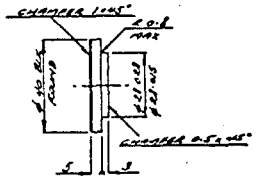
(15) PUSHROD
REQD:- 2 OFF
MATL:- Φ16 MS ROUND



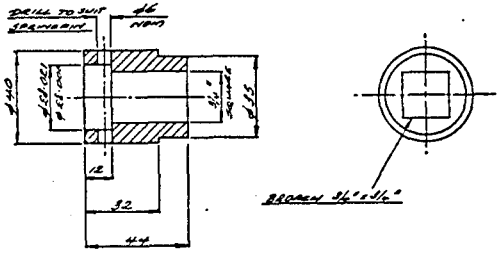
(4) (9) BEARING SUPPORT PLATE
REQD:- 4 OFF
MATL:- 5mm MS PL



(17) REAR OUTER BEARING HOUSING
REQD:- 2 OFF
MATL:- Φ75 MS BLK ROUND OR SUITABLE HOLLOW BAR



(13) HOUSING CAP
REQD 2 OFF
MATL Φ40 MS BLK RND



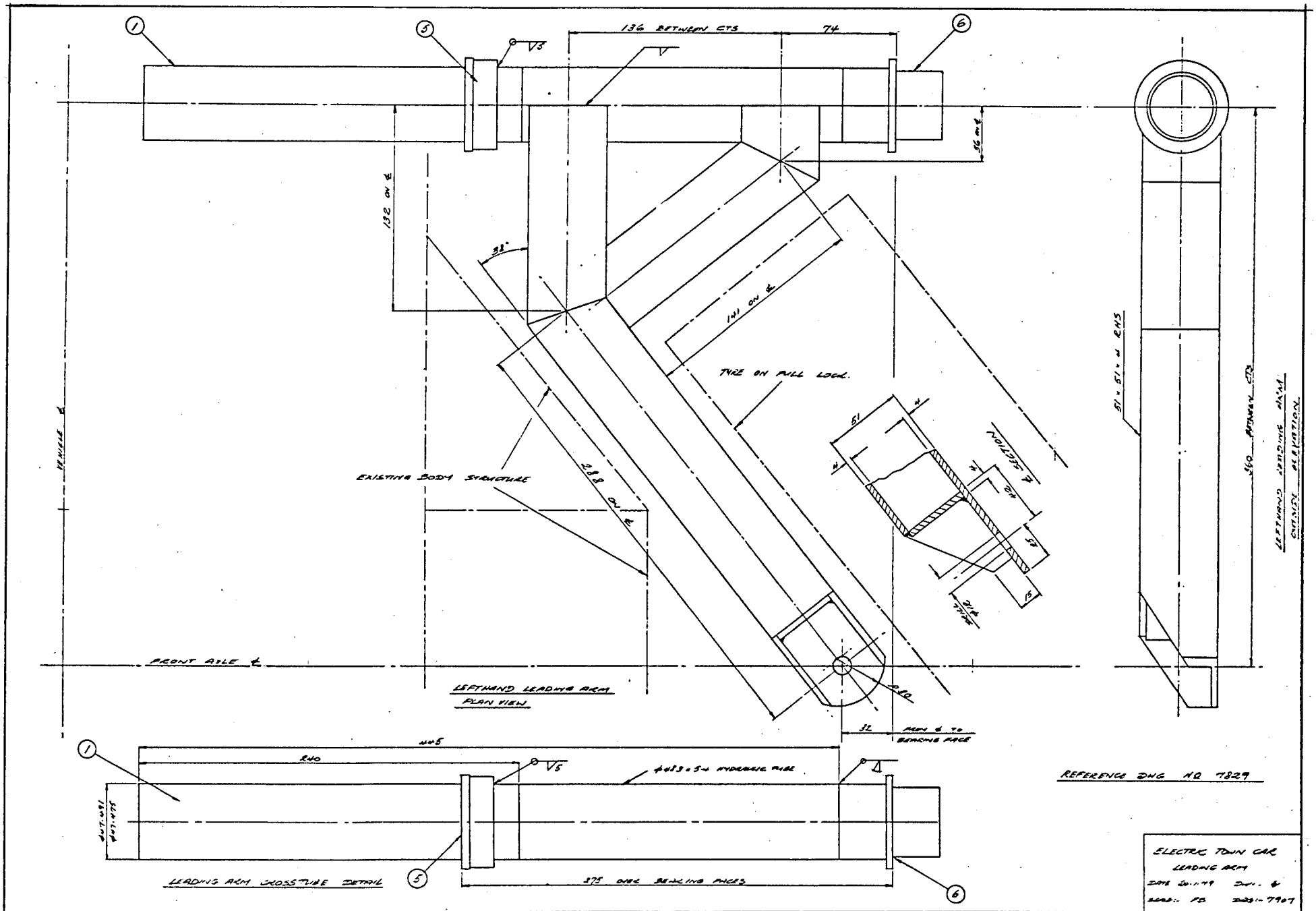
(19) OUTER HOUSING FOR AUXILIARY TORSION BAR
REQD: 2 OFF
MATL: Φ40 MS BLK ROUND

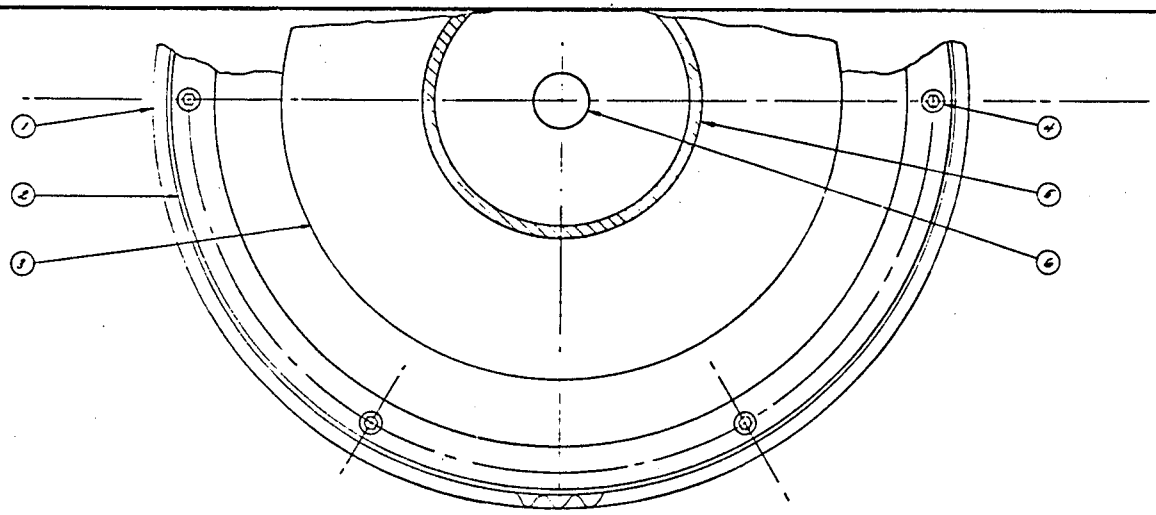
REAR - ITEMS 17, 13 & 19

ELECTRIC TOWN CAR
COMPONENT DETAILS

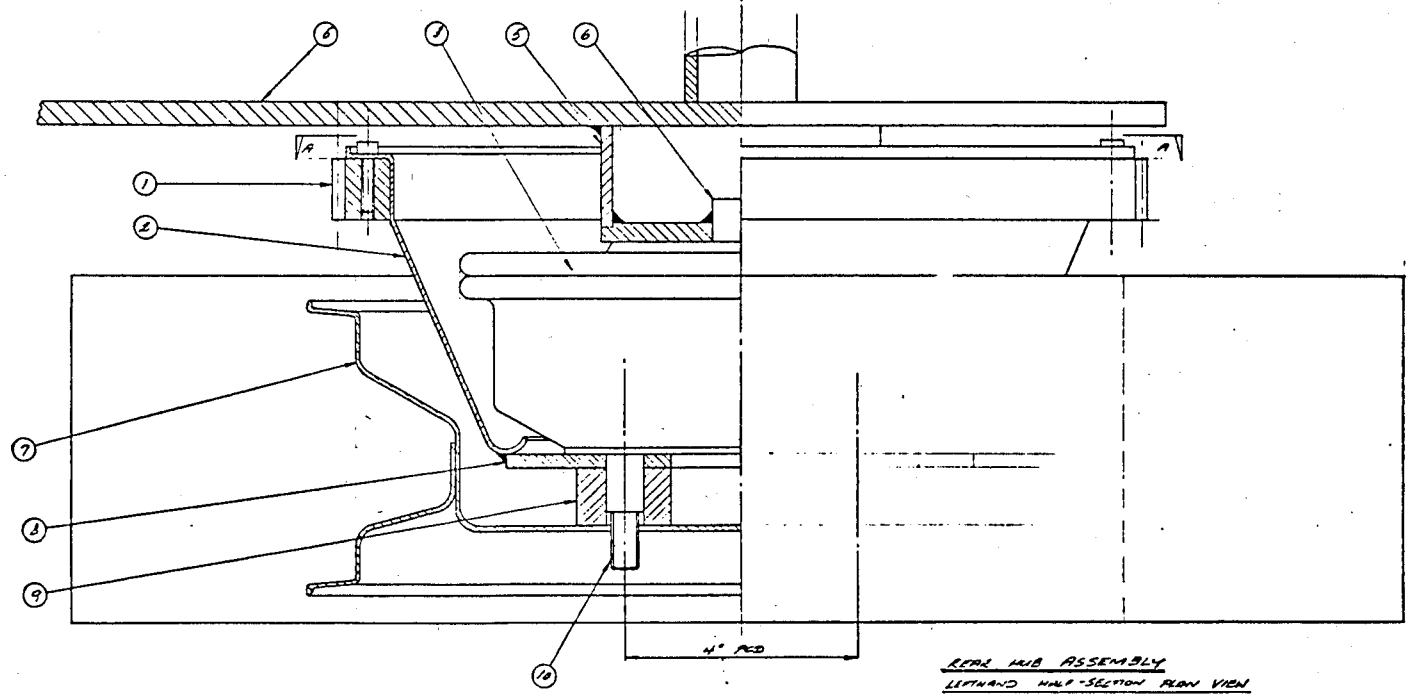
DATE:- 11-1-79 DRN- A

SIZE A3 ENR 7906/19


136 BETWEEN CTS
74
V3
V
1
5
6
132 ON ℄
38°
141 ON ℄
TYRE ON FULL LOCK.
EXISTING BODY STRUCTURE
288 ON ℄
WHEEL ℄
FRONT AXLE ℄
SECTION
R10
LEFTHAND LEADING ARM
PLAN VIEW
32
MEAN ℄ TO BEARING FACE
R40
V5
4x13 x 5w HYDRAULIC PIPE
SLOT 18W x 44L
LEADING ARM CROSSTUBE DETAIL
375 ONE BEARING FACES
51 x 51 x 4 RHS
360 RADIAL CTR
LEFTHAND LEADING ARM
OUTSIDE ELEVATION
REFERENCE DWG NO 7829
ELECTRIC TOWN CAR
LEADING ARM
DWG. ℄
7827



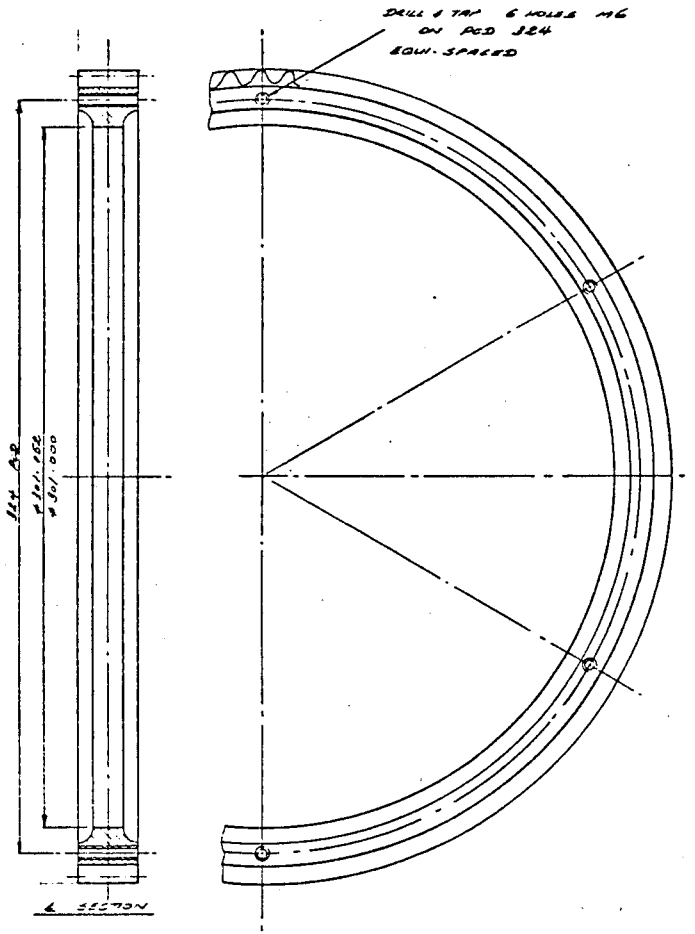
SECTION A-A



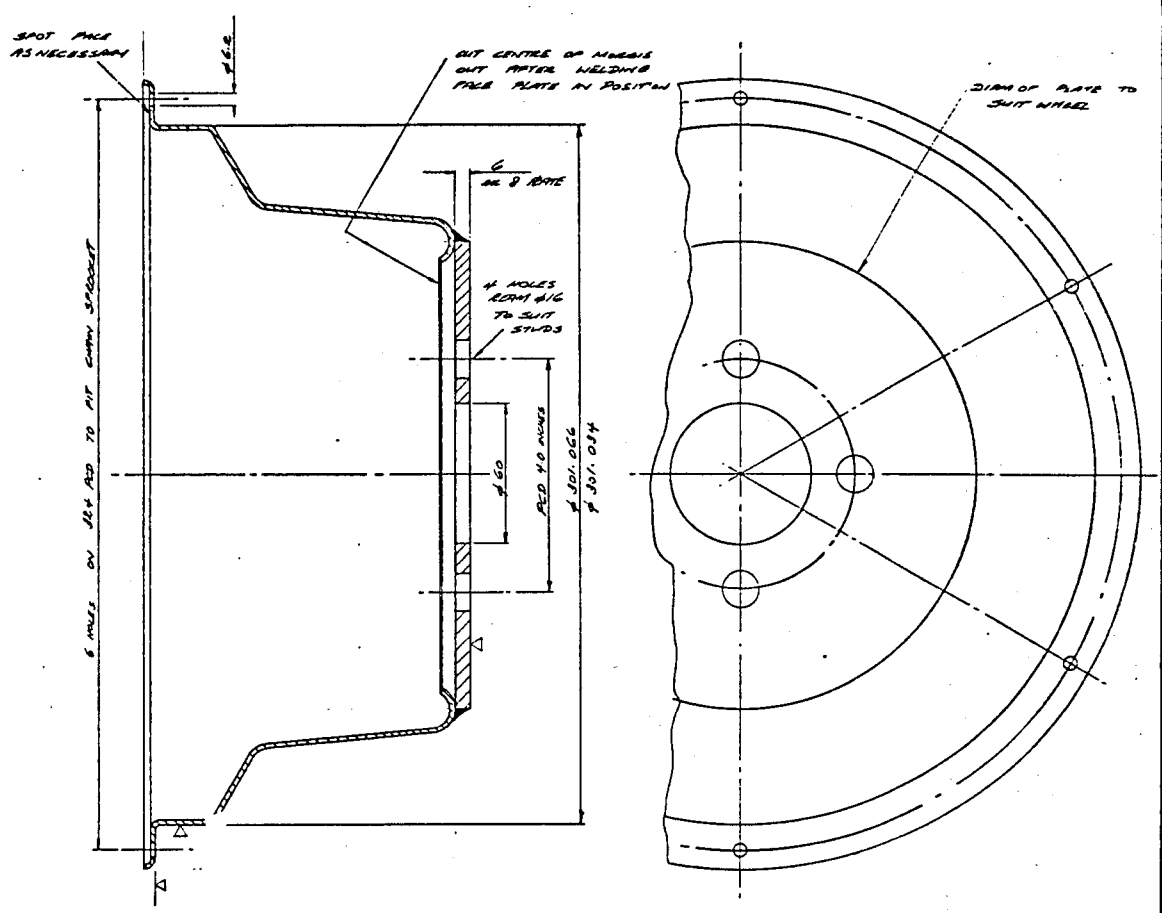
REAR HUB ASSEMBLY
LEFT HAND HALF-SECTION PLAN VIEW
CHAIN GUARD NOT SHOWN

ITEM	DESCRIPTION	REQ'D
1	3.00-4 CHAIN WHEEL	2
2	WHEEL FLANGE (27 HOLES)	2
3	WHEEL FLANGE (33 HOLES)	2
4	M8 x 25 LONG STUD BOLT	12
5	LEFT SUSPENSION ARM	2
6	AXLE ARM	2
7	4.5" HUB SHELL (ALUM)	2
8	FACE RING FOR HUB SHELL	2
9	25mm ALUMINIUM SPACER	2
10	LONG WHEEL STUDS (4.5" HUB)	8

ELECTRIC TOWN CAR
REAR HUB ASSY.
DATE: 21/1/79 DRN. W
SCALE FS DWG 7009

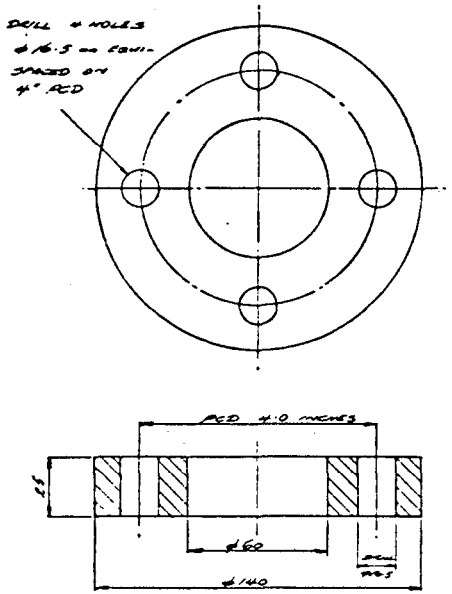


① 3-Inch-3 CHAIN SPROCKET
EF32 2 OFF

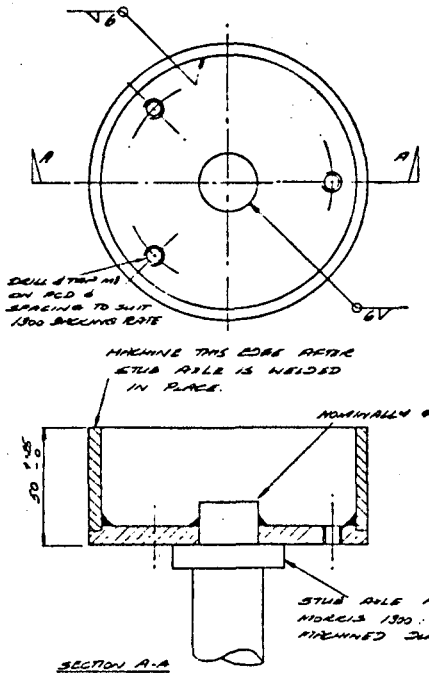


② ⑧ NORRIS/RISDON 1100 WHEEL AND FACE PLATE DETAIL
REQD: 2 OFF
MATL: FACE PLATE ex 8 PLATE

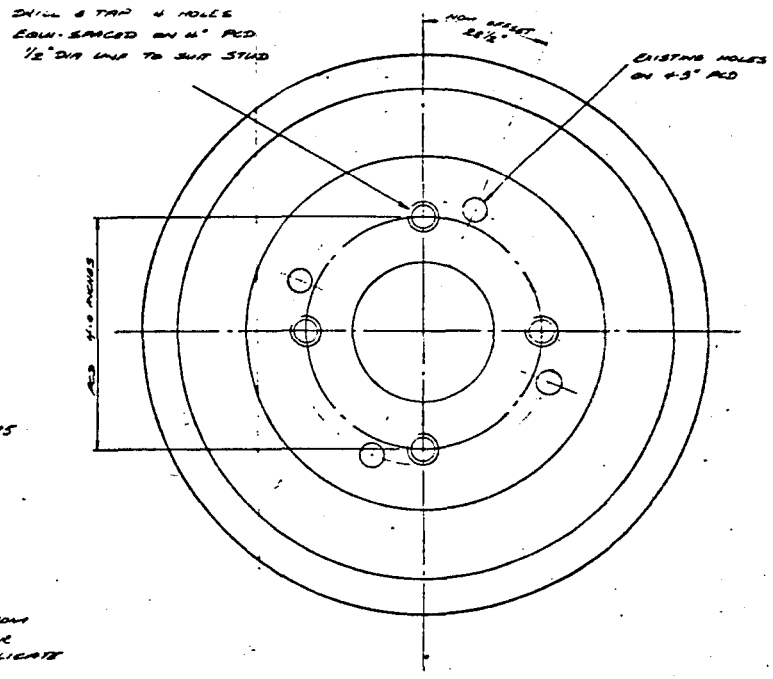
ELECTRIC TOWN CAR
CHAIN WHEEL DETAILS
DRG: 6/8/79 SHT: 8
SCALE: FS DWG: 7911



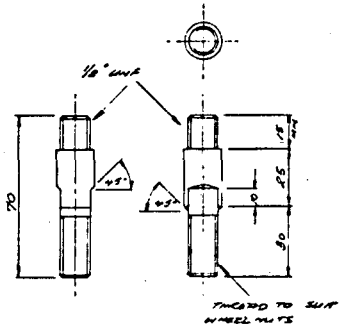
④ 25 mm ALUMINIUM SPACER
REQD: 2 OFF
MATL: 25 mm ALUMINIUM PLATE



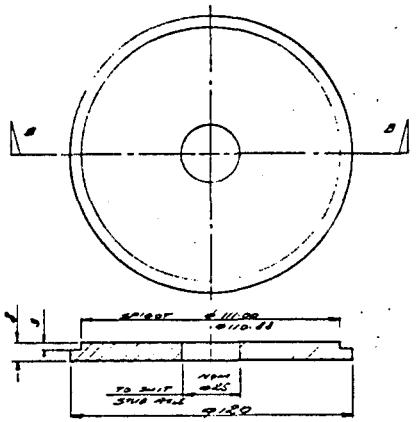
⑤ LINK SUSPENSION FROM STUB AXLE EXTENSION
REQD: 4 OFF
⑥ MATL SEE COMPONENT DRAWING BELOW



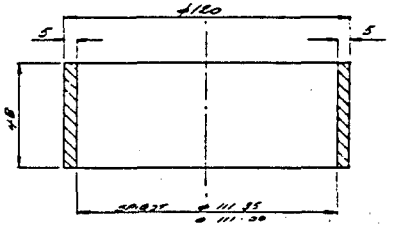
③ DRILLING DETAILS FOR MORRIS 1300
FOR SCALE N/B
REQD: 2 OFF



⑩ WHEEL STUDS
REQD 8 OFF
MATL: Ø16 BRIGHT

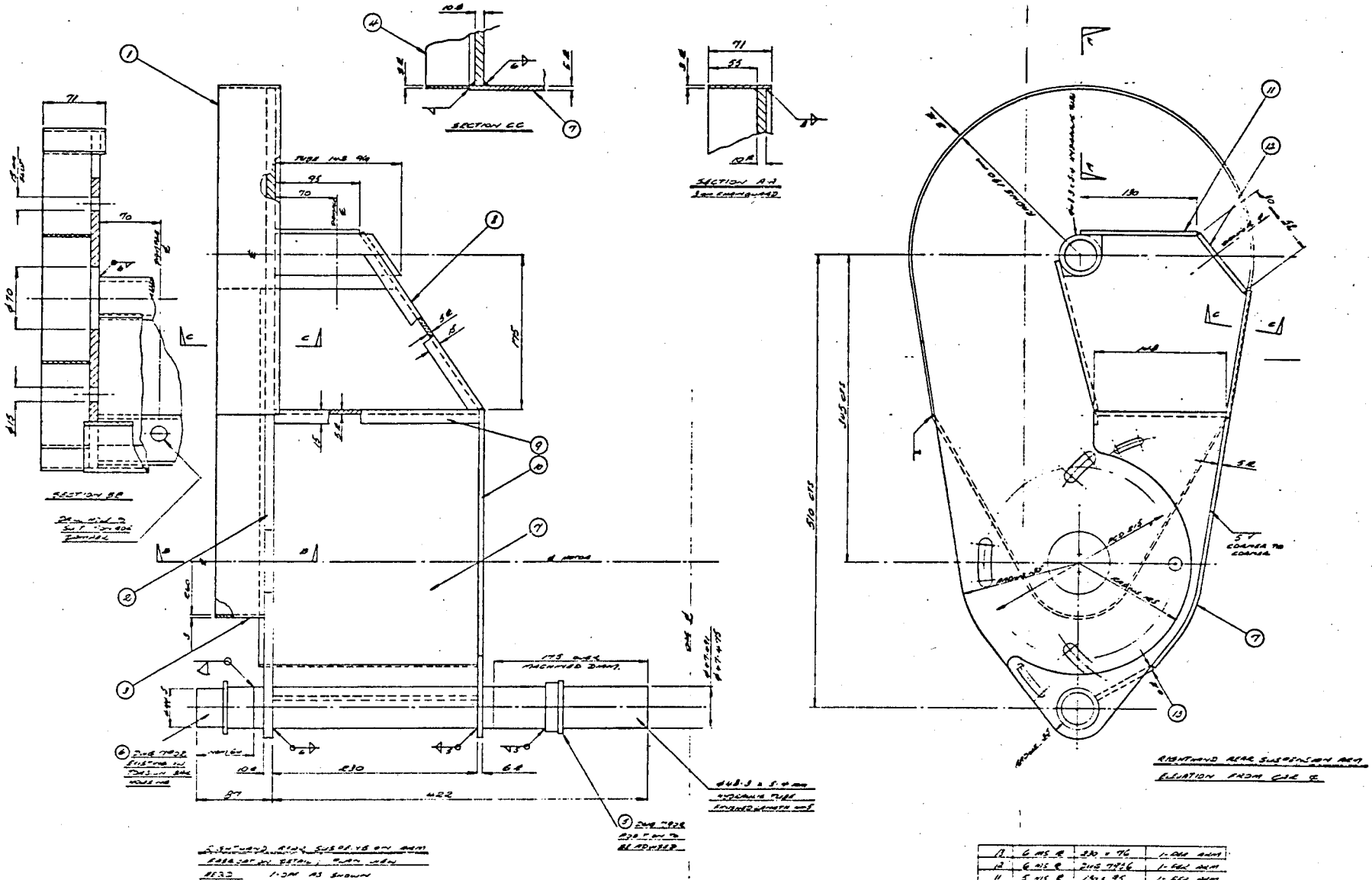


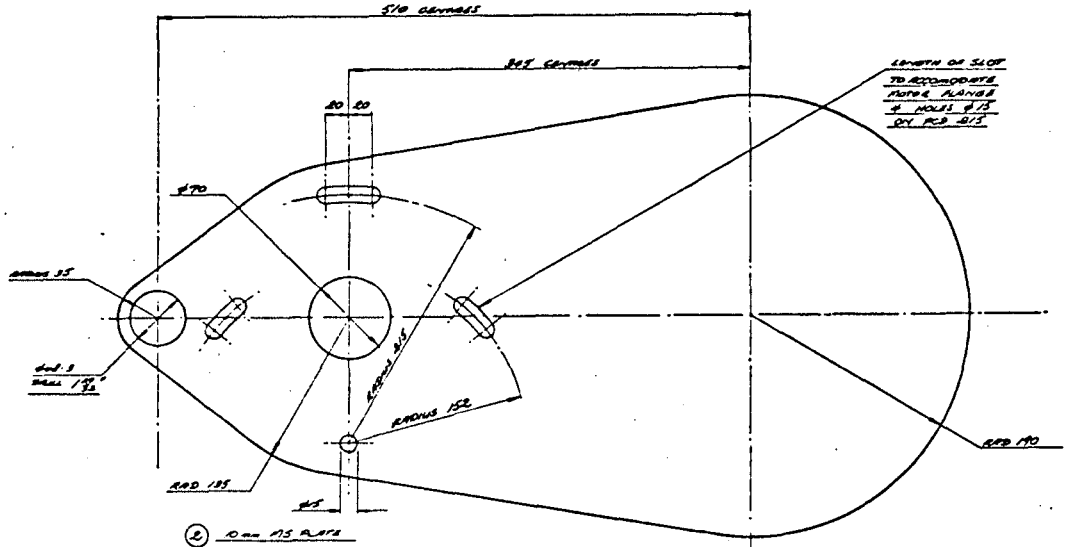
E SECTION BB. STUB AXLE EXTENSION DETAIL
REQD 2 OFF
MATL 3 mm MS PLATE



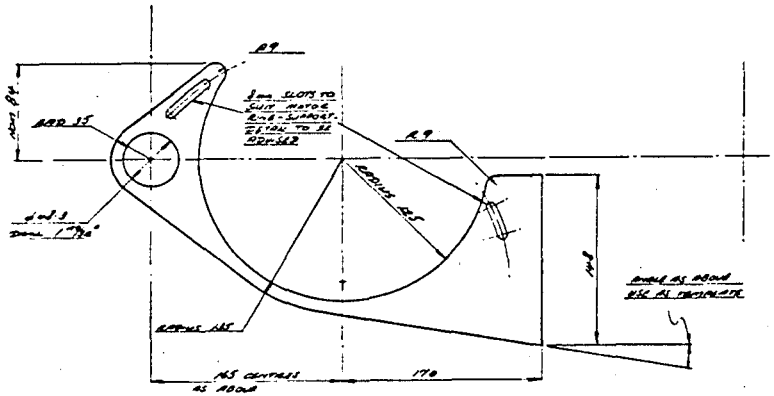
E SECTION: STUB AXLE EXTENSION DETAIL
REQD: 2 OFF
MATL: ROLLED 3 mm MS PLATE OR
SUITABLE PIPE SECTION

ELECTRIC TOWN CAR
REAR HUB COMPONENTS
DATE 6/12/79 DRN. H
SCALE FS DWG 79/2

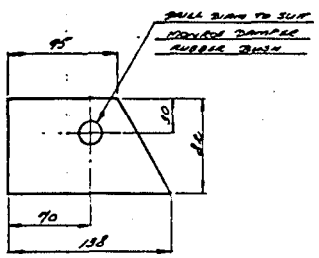

SECTION CC
SECTION AA
SECTION BB
RIGHT HAND REAR SUSPENSION ARM
ELEVATION FROM CAR C/L



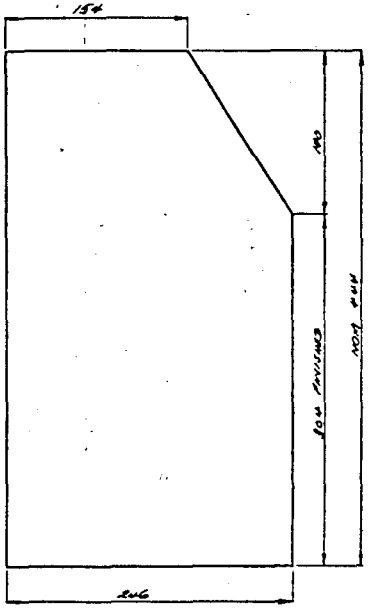
② 10mm MS PLATE
REQ'D: 1-OFF PER ARM



⑩ 6mm MS PLATE
REQ'D: 1-OFF PER ARM



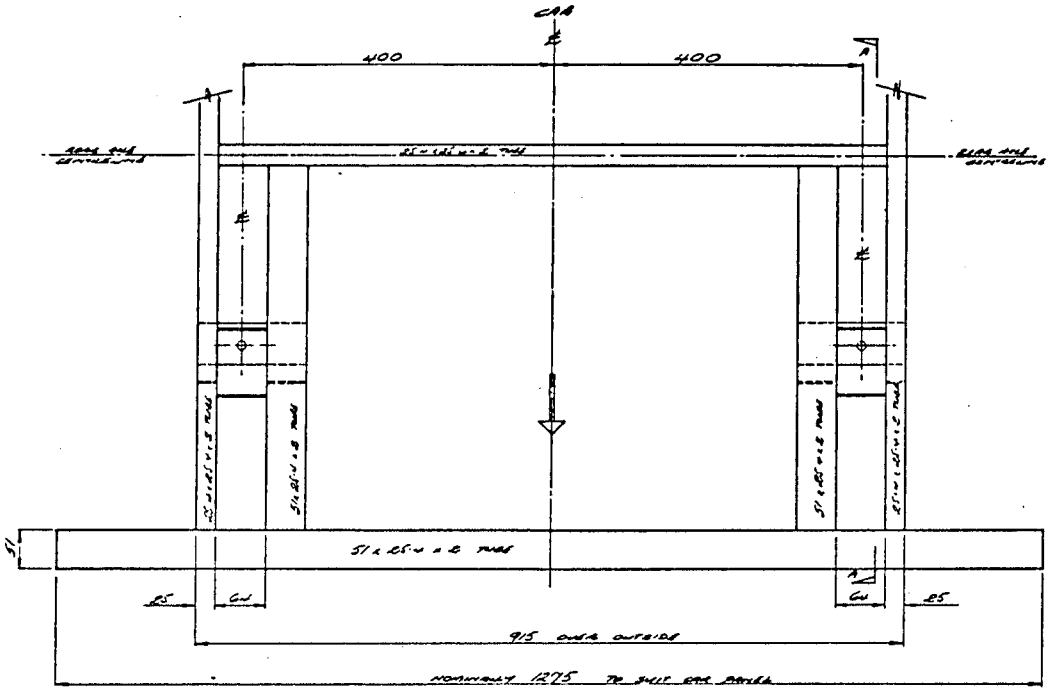
⑫ 6mm MS PLATE
REQ'D 1-OFF PER ARM



⑦ 5mm MS PLATE
REQ'D 1-OFF PER ARM

ELECTRIC TOWN CAR REAR DWG 7997

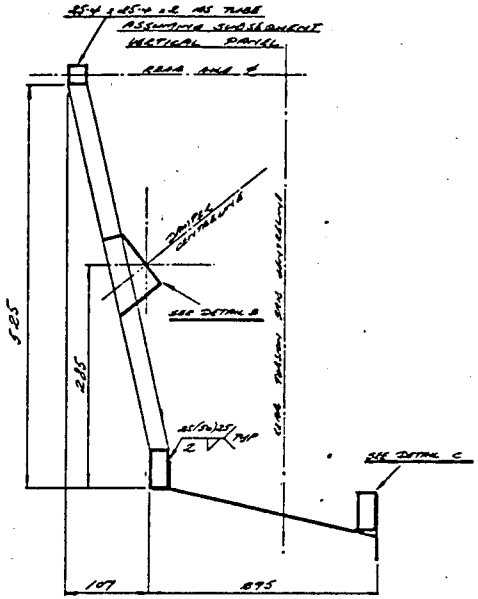
ELECTRIC TOWN CAR
SELECTED CUTTING DETAIL
FOR REAR SUSP. ARM
DATE: 4/9/99 DWN: ✓
SCALE: 1:8 DRG: 7956



PLAN VIEW

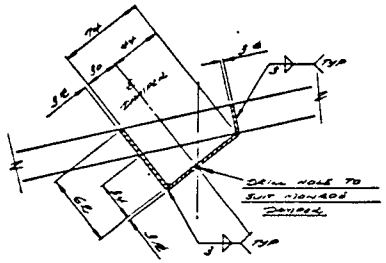
REAR BODY & DAMPER SUPPORT

N.T.S.



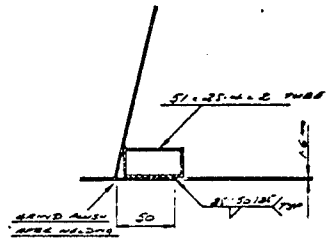
SECTION A-A

N.T.S.



DETAIL B

1:2

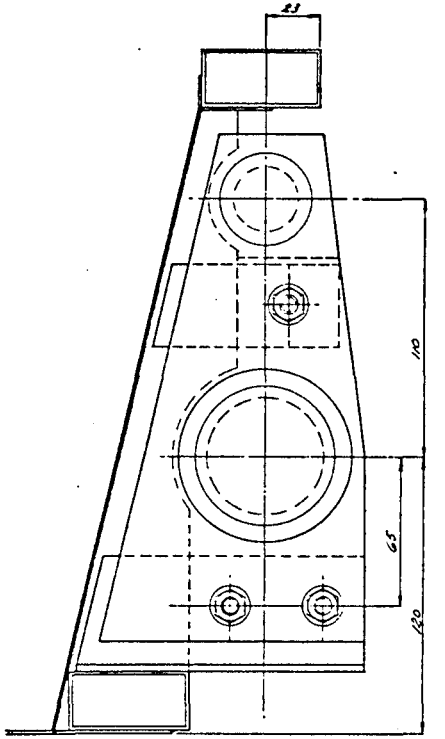


DETAIL C

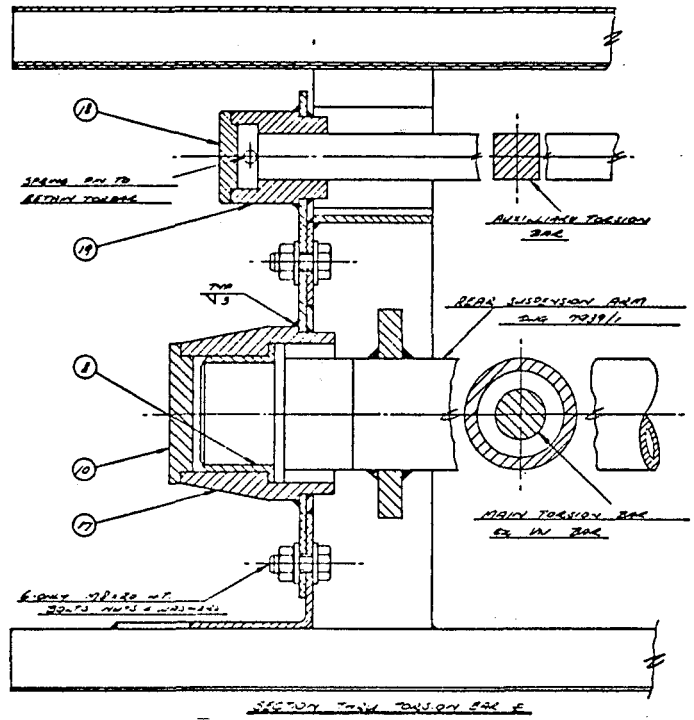
1:2

ELECTRIC TOWN CAR	
REAR BODY & DAMPER SUPPORT	
12/12/30	DWG #
SCALE - N.T.S	DWG - 3056

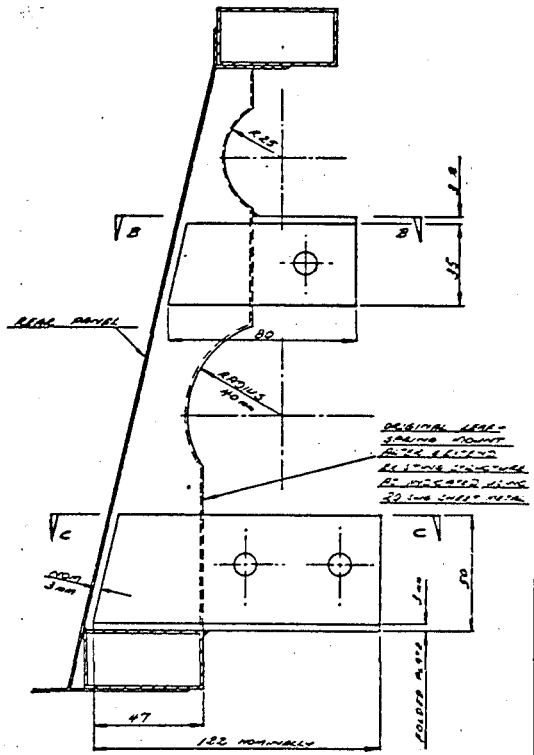
(?) DETAILS OF CALLED NUMBERS
SEE DWG 7902-A



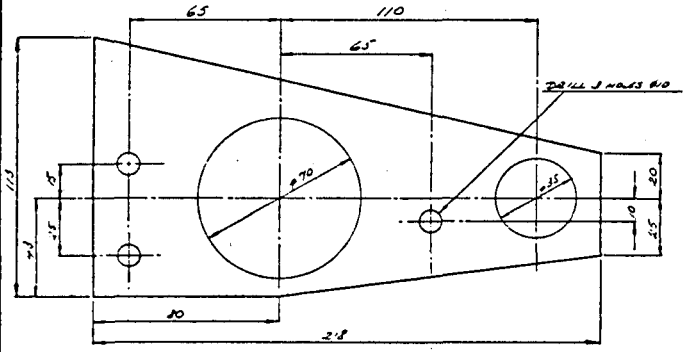
FIXING LOCATION OF REAR SUSPENSION PIVOT SUSPENSION ARM NOT SHOWN



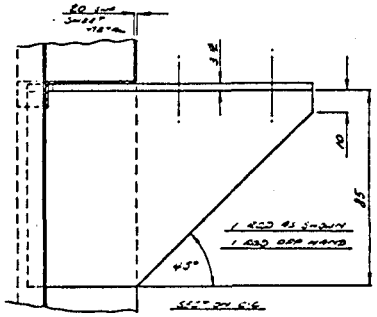
SECTION THRU TORSION BAR & REAR HAND REAR PIVOT OUTBOARD END IN SPHERE HOUSE



LOCAL STRENGTHENING ON BODYWORK AND MOUNTING WEB LEAF HAND SIDE

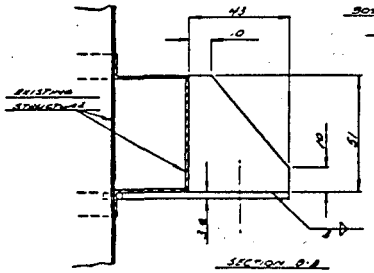


REAR SUSPENSION PIVOT PLATE
2 OFF 3 mm MILD PLATE



SECTION B-B

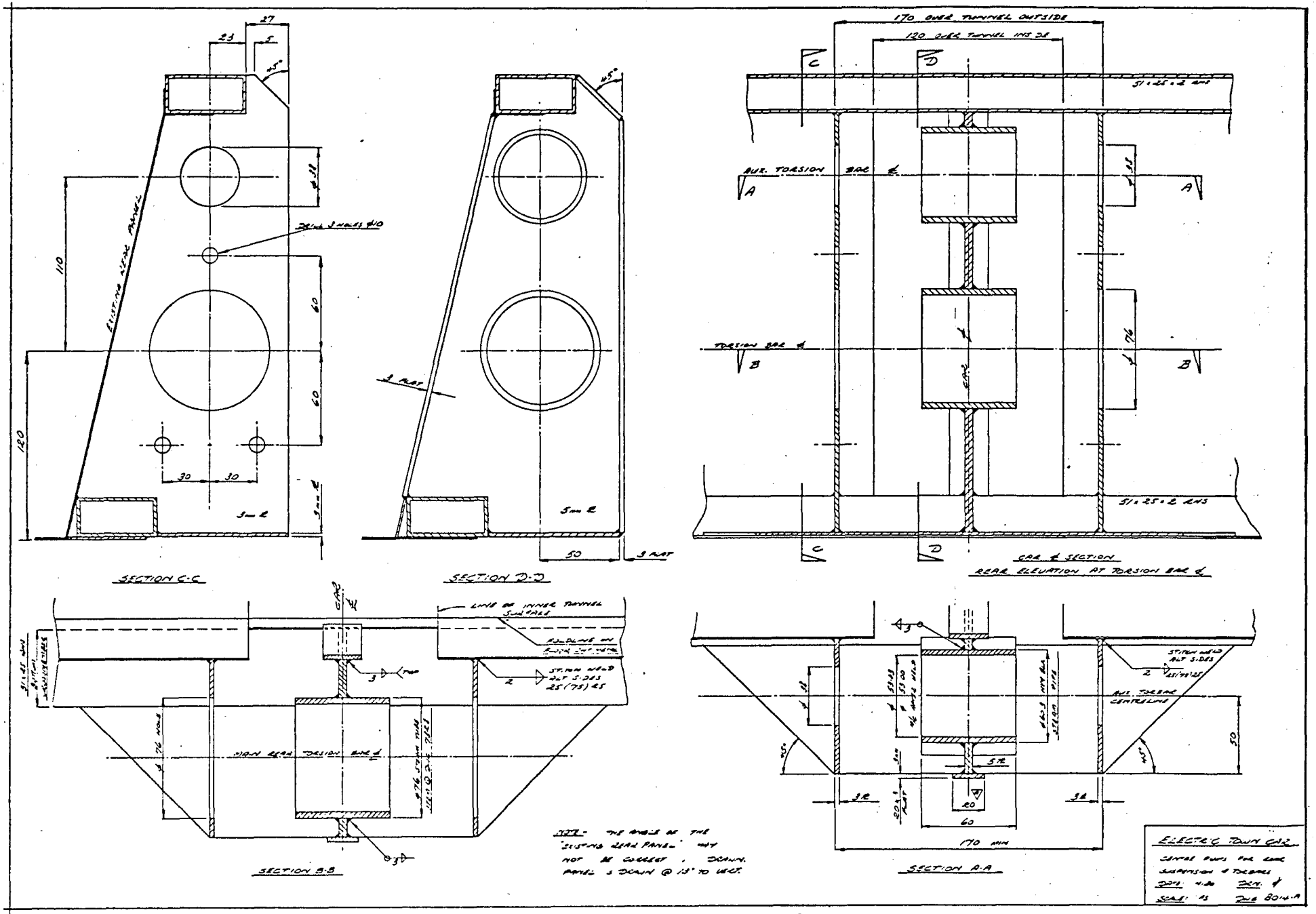
REAR MOUNTING PLATE
MILD STEEL 10 mm PLATE

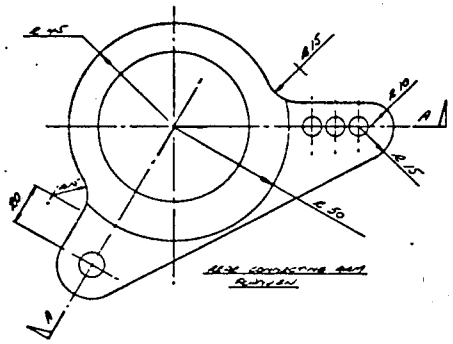


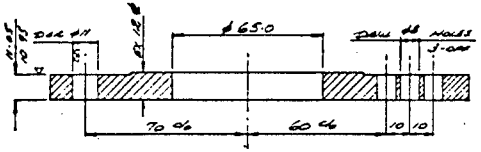
REAR MOUNTING PLATE
10 mm MILD PLATE
1 OFF RH HAND
1 OFF OPPOSITE HAND

NOTE: THIS DRAWING TO BE READ IN CONJUNCTION WITH DWG 8006 SEE ALSO DWG 7902-A AND 8014

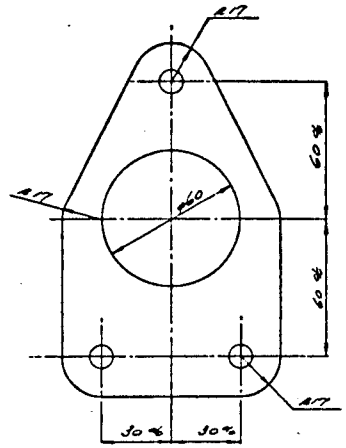
ELECTRIC TOWN CAR
DUAL REAR SUSPENSION
PIVOT BRACKET
DATE 7/80 DRY 4
SCALE AS SHOWN DWG 8013


SECTION C-C
SECTION D-D
CAR & SECTION
REAR ELEVATION AT TORSION BAR ℄
SECTION B-B
SECTION A-A
NOTE- THE HOLES IN THE 'EXISTING REAR PANEL' MAY NOT BE CORRECT, DRAWN PANEL IS DRAWN @ 13" TO U.C.F.
ELECTRIC TOWN CAR
CENTRE PINS FOR REAR SUSPENSION & TORQUES

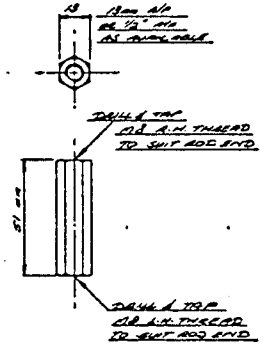




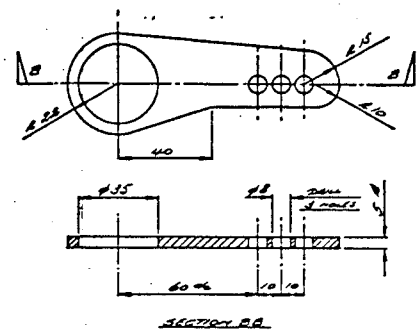
① REAR CONNECTING ARM
SECTION A-A
REQD:- 1-0M AS SHOWN
1-0M OPPOSITE HAND
MATL:- 12mm MS PLATE



② BACKING PLATE FOR LOWER
SUSPENSION ARM INNER PIVOT
REQD:- 2-0M
MATL:- 3mm MS PLATE

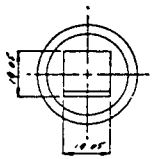


③ CONNECTING TIE ROD
REQD:- 2-0M
MATL:- MILD STEEL

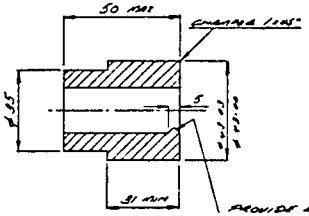


SECTION BB

④ CONNECTING ARM TO
EQUILIBRIUM TONGUE
REQD:- 2-0M
MATL:- 5mm MS PLATE

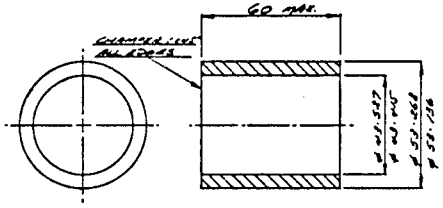


BROACH 1/4" & 1/8"
AT RACE END TO
PROVIDE LIGHT
PUSH FIT FOR
TORSION BAR.
NOTE BROACH AFTER
WELDING TO ARM ⑤

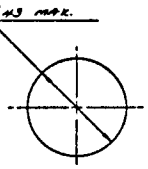


PROVIDE LUG
ON ONE INSIDE
FACE AFTER
BROACHING TO
PREVENT SLIDING
ALONG TORSION BAR
(A RUN OF WELD WILL
SUFFICE). GRIND FLAT
ON BAR AS DRG BOLT

⑤ EQUILIBRIUM TORSION BAR SPINDLE
REQD:- 2-0M
MATL:- ϕ50 MS BAR END



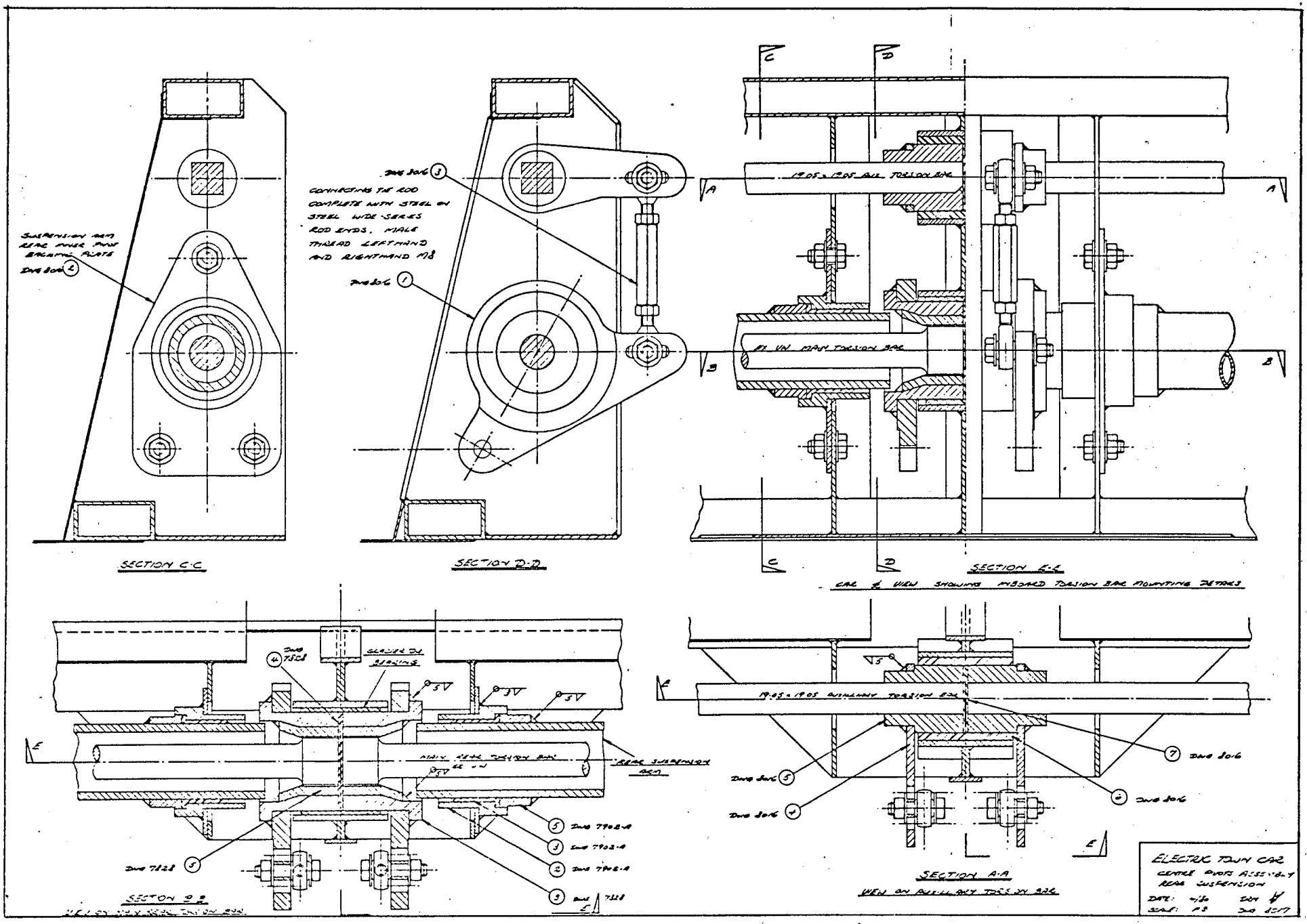
⑥ ANTI TORSION BAR BUSH
REQD:- 1-0M
MATL:- CELCON M90
BEARING MATERIAL
(CADILLAC PLASTICS)

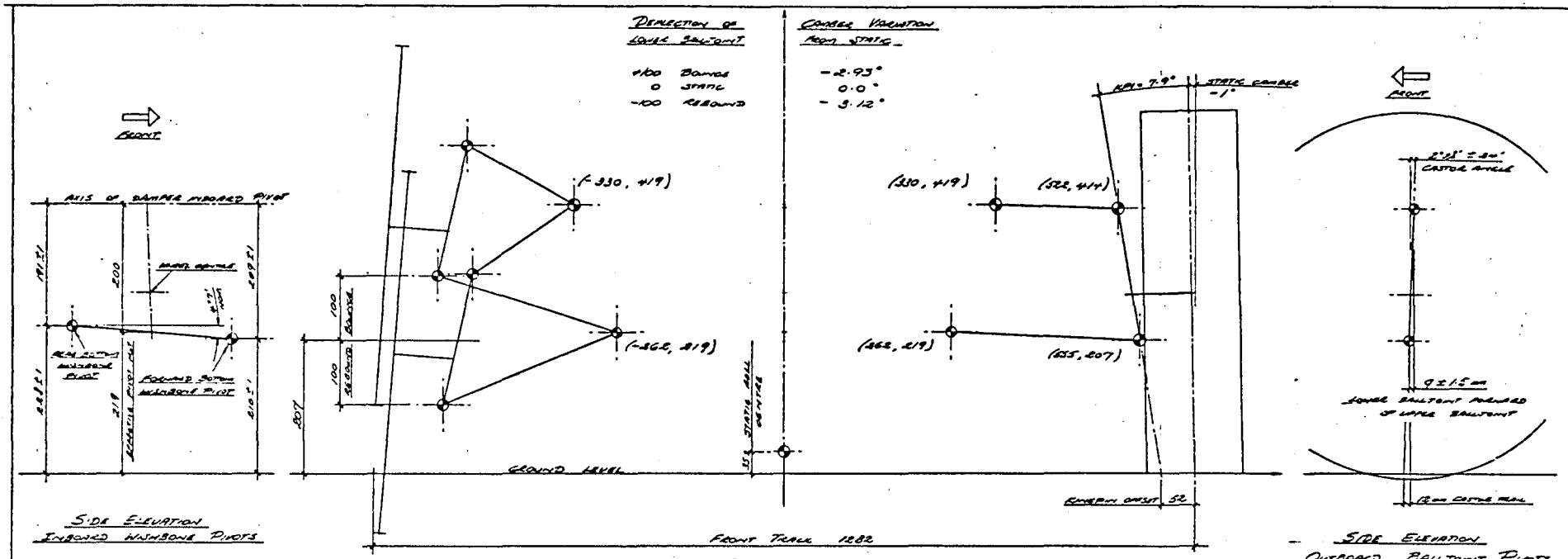


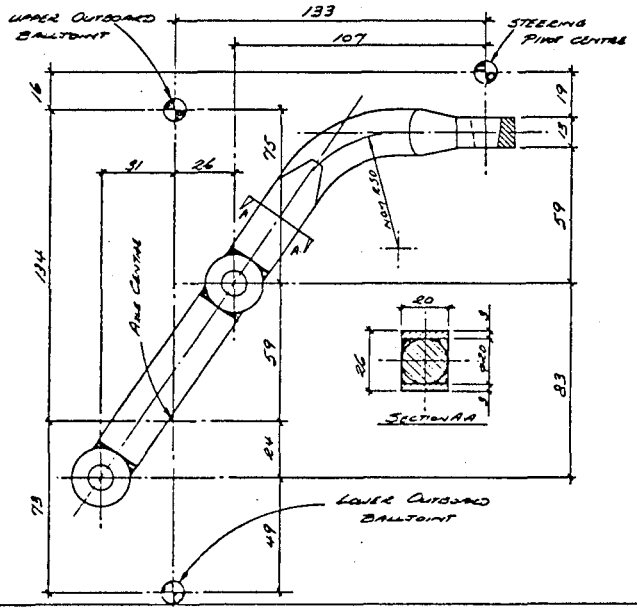
⑦ SEPERATING DISC
REQD:- 1-0M
MATL:- 3mm POLYETHYLENE

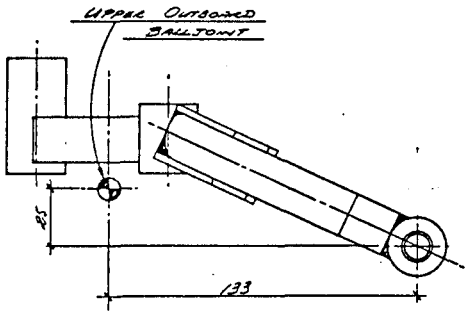
TO BE READ WITH 8017

ELECTRIC TOWN CAR
COMPONENTS FOR REAR
TORSION BAR PIVOTS
DATE:- 4/80 DRN:- A
SCALE:- AS DWG:- 8046


SUSPENSION ARM REAR INNER PIVOT BRACKING PLATE
Dwg No 6
SECTION C-C
CONNECTING TIE ROD COMPLETE WITH STEEL OR STEEL WIRE SERIES ROD ENDS, MALE THREAD LEFTHAND AND RIGHTHAND M8
Dwg No 6
Dwg No 6
SECTION D-D
19.05 x 19.05 AUX. TORSION BAR
IN VIN MAIN TORSION BAR
SECTION E-E
CAR ¼ VIEW SHOWING INBOARD TORSION BAR MOUNTING DETAILS
Dwg No 7501
SECURE BY BRAZING
5V
MAIN REAR TORSION BAR
REAR SUSPENSION ARM
Dwg 7928
Dwg 7902-A
Dwg 7903-A
Dwg 7906-A
7418
SECTION B-B
19.05 x 19.05 AUXILIARY TORSION BAR
Dwg No 6
Dwg No 6
Dwg No 6
Dwg No 6
SECTION A-A
WHEN ON AUXILIARY TORSION BAR
ELECTRIC TOWN CAR
CENTRE PIVOT ASSY. & REAR SUSPENSION
DATE: 4/60
SCALE: FS
DRN
DWG NO 7

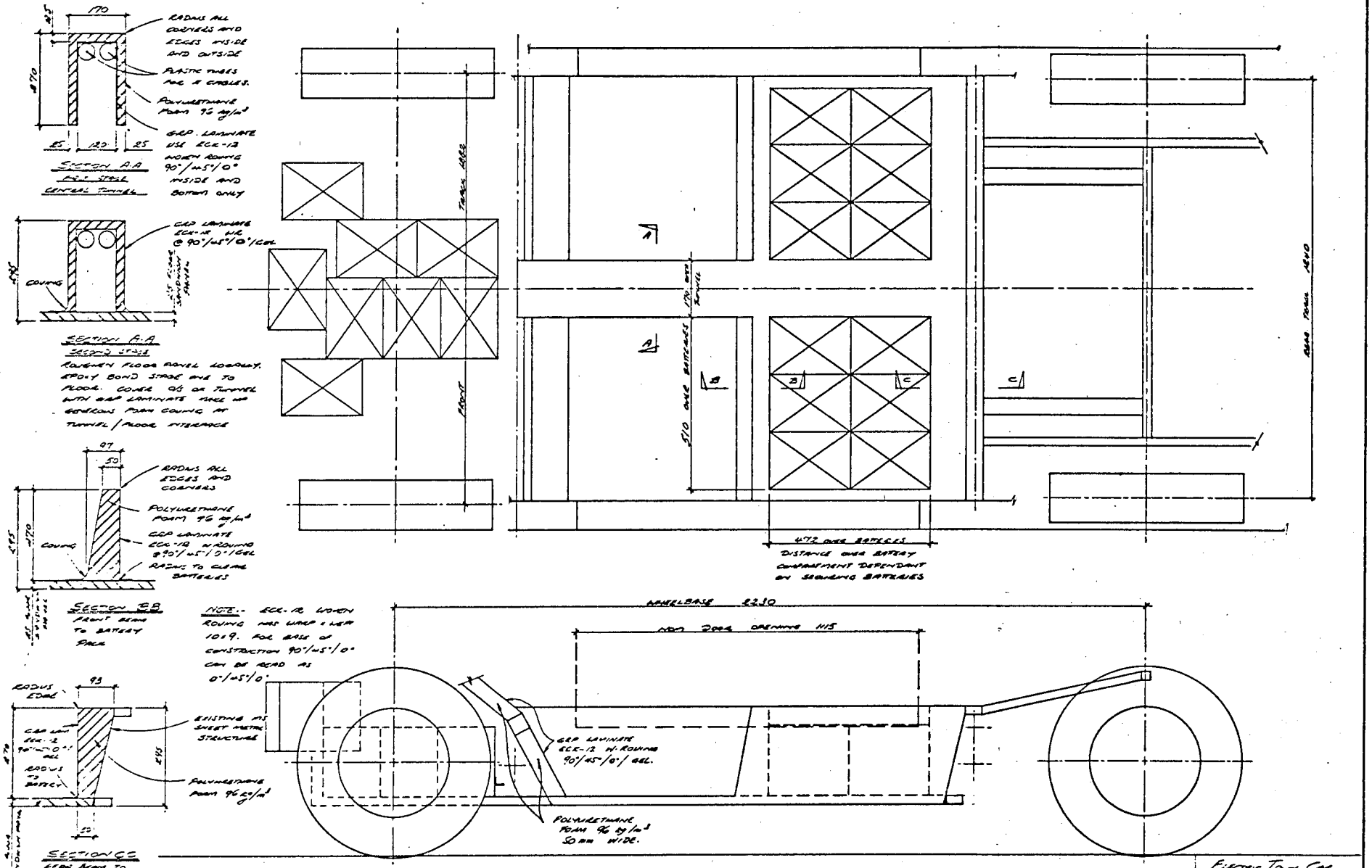






STEERING ARM DETAIL
PLAN & ELEVATION
TO SUIT RACK & PINION STEERING

ELECTRIC TOWN CAR
SUSPENSION & STEERING ARM DETAIL
DWG: 6/90 SHT: 4/
SCALE: 1:3 1/2 DATE: 2023


170
RADIUS ALL CORNERS AND EDGES INSIDE AND OUTSIDE
PLASTIC TUBES ARE A CIRCLES.
POLYURETHANE FOAM 96 kg/m³
GRP. LAMINATE USE ECK-13 WOVEN ROVING 90°/±45°/0° INSIDE AND BOTTOM ONLY
85
120
25
SECTION A-A
ALL STEEL CENTRAL TUNNEL
GRP LAMINATE ECK-12 H.L @ 90°/±45°/0°/GEL
COVING
SECTION A-A
SECOND STAGE
ROUGHEN FLOOR PANEL LOCALLY. EPOXY BOND STAGE ONE TO FLOOR. COVER OF TUNNEL WITH GRP LAMINATE TAKE UP GORDON FOAM COVING AT TUNNEL / FLOOR INTERFACE
97
50
RADIUS ALL EDGES AND CORNERS
POLYURETHANE FOAM 96 kg/m³
GRP LAMINATE ECK-12 H ROVING @ 90°/±45°/0°/GEL
RADIUS TO CLEAR BATTERIES
COVING
SECTION B-B
FRONT BEAM TO BATTERY PACK
NOTE:- ECK-12 WOVEN ROVING HAS WARP & WEFT 10:9. FOR BASIS OF CONSTRUCTION 90°/±45°/0° CAN BE READ AS 0°/±45°/0°
RADIUS EDGE
93
EXISTING M.S. SHEET METAL STRUCTURE
GRP LAMINATE ECK-12 H. ROVING 90°/45°/0°/ GEL.
RADIUS TO BATTERY
POLYURETHANE FOAM 96 kg/m³
50
SECTION C-C
REAR BEAM TO BATTERY PACK AND SUSPENSION SUPPORT BEAM
TRACK GAUGE
FRONT
A
B
C
510 OVER BATTERIES
472 OVER BATTERIES
DISTANCE OVER BATTERY COMPARTMENT DEPENDANT ON SEQUENCING BATTERIES
WHEELBASE 2230
NON DRIVE OPENING 415
POLYURETHANE FOAM 96 kg/m³ 50 mm WIDE.
ELECTRIC TOW CAR
CENTRAL FRAME AND BATTERY STRUCTURE
DWG. 7/6 DON
SCALE - 1:5 DWG NO 4